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Learning by Rote

THERE are minds that are so constituted that they must do everything by rote. They commit rules by heart and go by them. They get directions and follow them, as well as they can remember. They have to have particular instructions for each step in the proceedings and each corner that they turn.

There are other minds that seem to grasp the thing in its entirety, to lay hold of the principle of it, to sense the lay of the land and know how to get to where they are going.

One man will learn a line of rules about the lever, levers of the first order, levers of the second order, etc., and if he does not get them mixed up and misapply them, he will figure out what a given lever may be expected to do with a given force applied to it.

Another man with less mental strain will learn the principle of moments and be forever after able to make his own rules, analyze systems of levers, simple and compound, and recognize them in the pulley and other mechanisms.

One man will get lost in the woods of mechanical application. He will confuse force with energy and get an idea that he can produce or multiply power by mechanism. A glance at the thing from one of the high spots of physics would show him that he was working up a blind alley.

Try to see a subject broadly, to get at its fundamentals and relationships. If you know that the sun is in the south and you ought to go east, that you should follow a railroad or a river, or keep a chain of mountains to your left, you will not be led far astray by faulty or misunderstood directions, and you won't have to ask for so many either.

Try to get the big broad idea of things, why if one is such the other must be so, rather than to learn a lot of things by rote, the reason for and often the application of which, you do not understand.

F. R. Low

The Use of Welding in Power-Plant Piping

By LEWIS J. SFORZINI

WELDED pipe lines—but recently a curiosity—are now in use and under construction in power plants all over the country. The author, who has made extensive use of welded piping installations for a variety of purposes, here gives the essence of his experience for the benefit of engineers not yet fully acquainted with the time- and money-saving developments in this field

THE rapid development of economical methods of welding by both the oxyacetylene and the electric processes, which began near the advent of the world war, has brought these forms of welding into extensive use in the power-piping industry. Today

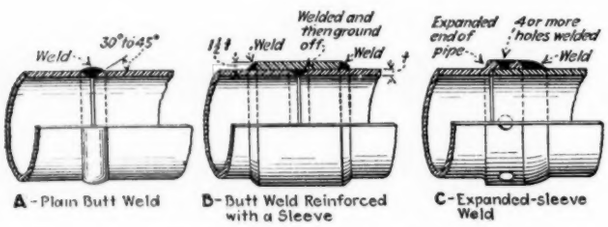


FIG. 1—TYPES OF LINE WELD USED TO REPLACE FLANGES

there is hardly a single pipe-fabricating concern that does not employ this type of welding in some way or other in the fabrication of its product. Besides its employment by the fabricators, it is used as an economical or expedient method by the power plant itself in making alterations and extensions to existing piping systems. It is the purpose of this article to indicate how the piping designer or the power-plant operator can use welded piping to advantage for certain conditions.

In high-pressure piping systems, in which class may be placed all those conveying steam, water, air, brine, ammonia and other fluids at pressures above 30 lb., welding may be used to give compactness, to reduce the number of joints (and hence avoid some of the potential leaks), to minimize radiation losses by doing away with flange radiating area, to obtain lower first cost of piping under certain conditions, to obtain quicker delivery on pipe with welded outlets over that obtainable on special fittings and to reduce the cost of pipe covering under certain conditions, particularly in low-temperature work with brine or ammonia where cork is used. In addition to the foregoing the closing length of a pipe run can, where time is an important factor, be measured in the field and the pipe burned off and the ends welded together without the necessity of sending away to have the pipes flanged or threaded the proper length. The present trend toward higher steam pressures and temperatures is leading to the elimination of as many gaskets as possible and to the sealing of steel flanges by welding.

Of the foregoing advantages, the reduction of the radiation area by eliminating flanges is ordinarily of minor importance. With long pipe lines, however, this factor is worth considering. For instance, in a recently installed outdoor 12-in. high-pressure steam line about 5,000 ft. long, there were 250 welded joints. A pair of 12-in. extra-heavy flanges has a total radiating area about 3½ sq.ft. greater than that of plain pipe, with a loss of 100 B.t.u. per square foot per hour from covered areas, or 350 B.t.u. per pair of flanges. With steam continuously in the pipe, the saving by omitting flanges would be about 35 tons of coal per year. In addition, the necessity for expensive flange covers would be obviated.

Fig. 1 shows three of the common line welds used to avoid flanged joints on steel pipe. The simple butt weld *A* can be made either by oxyacetylene or electric

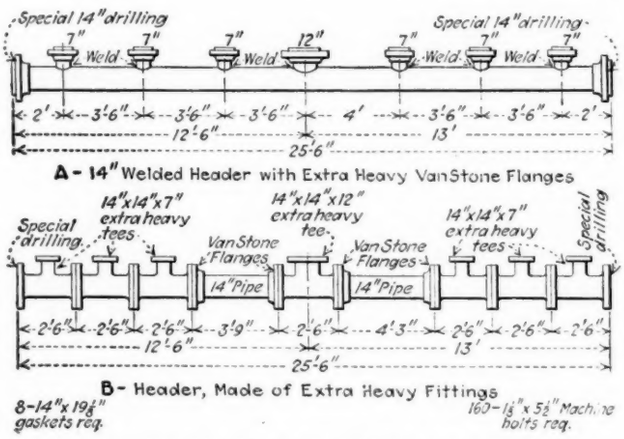


FIG. 2—THE USE OF *A* INSTEAD OF *B* SAVED $500

welding. In making this weld, the ends of the pipe are first scarfed at an angle of between 30 and 45 deg. The ends of the pipes are then brought together until they nearly touch, and aligned, after which a weld is made, working gradually around the periphery until the depression is completely filled. If this weld is made by a conscientious and expert operator under proper

supervision, it can be used successfully, as has been proved in numerous installations for steam at high pressures and superheats. However, some engineers are reluctant to use this joint without some added security against failure.

The combination butt-and-sleeve weld *B* can also be made up by either oxyacetylene or electric welding. The butt weld is made up as just described, after which the welded portion is ground off flush with the pipe by a portable grinder. A sleeve is then slipped over and welded at the ends as shown. The sleeve here acts as a reinforcement against rupture by pressure and failure by expansion stresses (in steam work). An experienced operator can weld a joint of this kind on a 16-in. pipe in about 3½ hours. If a sleeve is hard to procure in repair or alteration jobs, a screw pipe coupling with threads machined off, may be used.

The sleeve weld *C* is made by one fabricating concern, using the oxyacetylene process. About six inches of one end of each pipe is expanded by special machinery in the shop, after which four to six holes (depending on the pipe size) are burned through the expanded portion. In welding, the plain end of one pipe is slipped into the expanded portion of another and welded through the holes. The end of the expanded portion is then scarfed with the welding flame and a long weld made as shown. A rubber-manufacturing company has a 14-in. high-pressure steam line about

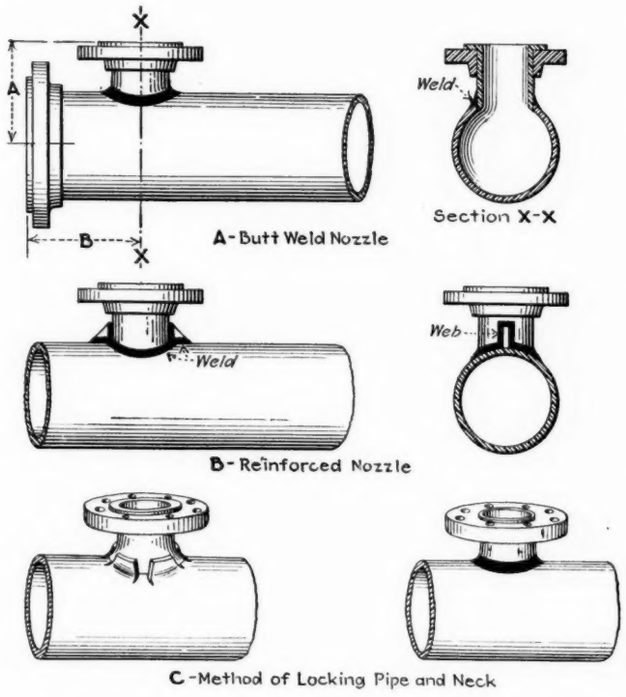


FIG. 3—VARIOUS METHODS OF ATTACHING NOZZLES

3,500 ft. long with this type of joint, and another manufacturing concern has a line with 250 such welds. An operator can make this weld on a 12-in. line in about 2½ hours. This joint, as well as those shown at *A* and *B*, can be welded from underneath when it is impossible to turn the pipe and weld from above.

In Fig. 2, *A* shows a 14-in. 250-lb. steam header from an actual installation. This was an alteration job where the 25 ft.-6 in. and 12 ft.-6 in. dimensions were fixed by existing construction. The end 14-in. flanges went against flanges used before the "American Standards" for drilling were adopted, and had to be specially drilled. Delivery on this particular job was important. The whole header was fabricated and shipped by the fabricator one week after receipt of order, at a small

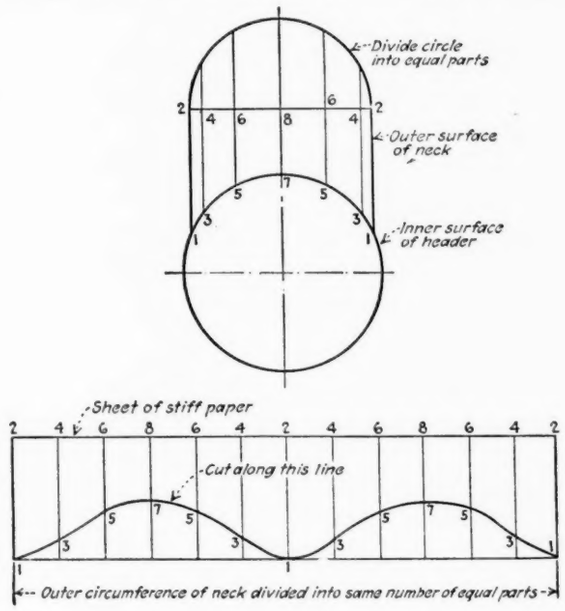

Divide circle into equal parts
Outer surface of neck
Inner surface of header
Sheet of stiff paper
Cut along this line
Outer circumference of neck divided into same number of equal parts

FIG. 4—METHOD OF LAYING OUT TEMPLET FOR CUTTING NOZZLE NECK TO FIT CURVE OF PIPE

fraction of the cost of the alternative construction *B* based on the use of fittings.

In forming outlets on headers, as shown at *A* of Fig. 2, various types of necks may be employed. Fig. 3 *A* shows a neck formed by the butt-weld process, the weld being of the same type as the butt line weld already described. *B* shows a weld reinforced by two webs, while *C* illustrates the formation of a neck by the "interlock" process.

Method of Forming Outlets on Headers

For small outlets on large pipe no curve is necessary on the necks. When outlets are almost equal in size to the headers, a templet can be made on stiff paper as shown in Fig. 4, which is self-explanatory. The templet is then placed on the neck, and the neck marked and burned with the cutting torch along the marked line.

All of the welds shown in Fig. 3 are made by piping fabricators in the shops with special facilities for doing this work economically and quickly, and in most cases a clean-cut job results, but in certain cases that came under the writer's observation, the welding material was allowed to flow to the inside of the pipe, leaving fins around the neck which caused an excessive pressure loss.

The flanged outlets welded on a header generally have the distance *A* (Fig. 3 *A*) equal to that of a standard fitting where standard flanges are used, or equal to the center-to-face dimension of an extra-heavy fitting where extra-heavy flanges are used. The distance *B* is limited only by mechanical clearances when near a Vanstone flange. Near a screwed flange, it should be at least 12 in. for small outlets, and the distance should be increased for the larger-sized outlets to prevent leaks

at the thread due to distortion of the metal in welding.

On headers with welded outlets it is advisable to use only cast or forged-steel flanges and never cast-iron flanges, because were the cast-iron flanges to break, the whole header would be useless without some makeshift arrangement. On rush alteration work, where a connection has to be made to an existing header during

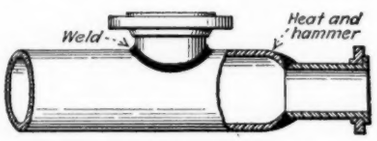


FIG. 5—METHOD OF JOINING TWO PIPES OF DIFFERENT SIZE

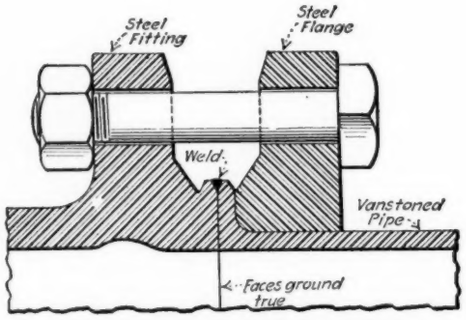


FIG. 6—JOINT USED FOR STEAM PRESSURES 350 LB. AND OVER

a short shutdown, a neck can easily be welded to the pipe right in place by using the neck as a templet and marking, then burning a hole and welding the neck in place. The necks for alteration work can be purchased or made from a piece of pipe with flange screwed on, by burning out a portion of the pipe to the proper contour.

Fig. 5 shows the method of welding a smaller to a larger pipe near an outlet. The cost involved in making this weld is justified only when a valve is to be placed against the flange, saving in either the cost of the smaller valve or in space, where clearance is needed.

Weld Used as Seal on High-Pressure Joint

Fig. 6 depicts a type of flanged joint that is becoming very common in high-pressure and high-temperature work; that is, over 350-lb. steam pressure and any temperature. This joint is made up of two Vanstone ends with special flanges or with two special steel fittings or the combination of a flange and a fitting. No gaskets are used, but the faces are ground true and smooth in the shop. These finished surfaces are brought together in the field and bolted tight, after which a sealing weld is applied. This joint does away with all gasket troubles by omitting the gasket. Further, since the bursting and tensile stresses are all carried on the bolts and not on the weld, the joint is efficient enough to meet the most stringent of safety requirements.

Another type of high-pressure joint used on steel fittings is shown in Fig. 7. This joint is used in steam or other high-pressure work when it is desired to avoid flange joints, and space requirements prohibit the use of a pipe bend. It is also a favorite for underground work where a gasket joint would be inaccessible for repairs.

On all ordinary piping, whether put up with welded, screwed or flanged joints, it is sometimes necessary to make small connections for outlets, for drips, for thermometer or gage connections, or for drains. Usually it is possible to tap fittings, valves or blind flanges for connections. For good work it is usual to limit the size to which a fitting is tapped to 1½ in. on standard fittings and 2 in. on extra-heavy fittings. The pipe itself may be tapped if the walls are sufficiently thick. A pipe-wall thickness of $\frac{7}{16}$ in. permits of tapping holes up to ½-in. pipe size; ¾-in. wall thickness permits pipe taps up to ¾-in.; with thinner walls or larger taps any of the welds shown in Fig. 8 may be used.

It is customary to test all welded, high-pressure piping at a hydrostatic pressure equal to twice the working pressure, after which all porous welds are made tight.

Testing Ammonia Lines With Air

For ammonia lines where there is even a remote possibility of the test water remaining in the pipe, water should not be used but the ammonia compressor should be used to pump air into the line until the pressure is equal to twice the probable maximum condenser pressure. Leaks can then be detected by loss of pressure when allowed to stand for some time, the exact location being determined by applying a soap and water solution with a brush. Air bubbles will quickly show the leak.

Until a very few years ago the designer of low-pressure steam and vacuum work, in using large mains (16-in. or over), had the choice of either cast-iron fittings or fittings made up of riveted steel plate. For pipe he also had the choice of either cast-iron pipe, which usually came in 12-ft. lengths with integral flanges, or steel pipe with riveted steel flanges on the ends. Cast-iron pipe and cast-iron fittings proved extremely heavy for certain installations, and the bolts used for making up the joints were also heavy from the standpoint of good design. In making up a run of

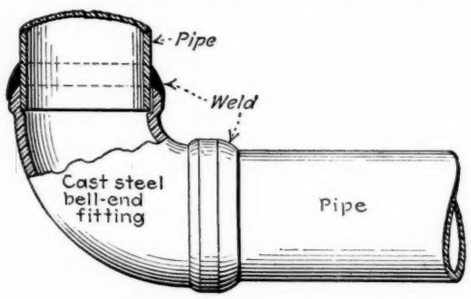


FIG. 7—HIGH-PRESSURE STEAM FITTING USED FOR TURNING SHARP CORNERS AND FOR UNDERGROUND WORK

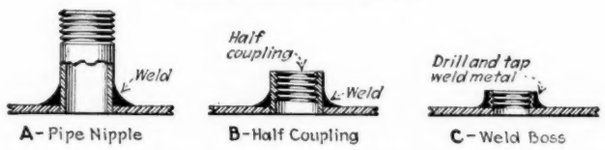


FIG. 8—METHODS OF MAKING SMALL CONNECTIONS TO PIPE WITH THIN WALLS

cast-iron pipe, either a special length of pipe had to be procured as a filler piece or a dutchman had to be used, either of which proved expensive.

Steel pipe with light riveted steel flanges and cast-iron fittings proved a better combination. However, there was still room for improvement, as it was apparent that there was no need for a pair of joints every time an outlet, no matter how small, was required. This brought about outlets riveted to the steel pipe by

the fabricators, and in some cases even riveted elbows and miscellaneous special fittings. The fabricator, in thus obtaining the whole order, was enabled to substitute lighter flanges than the American Standard in the form of the riveted-pipe-manufacturer's standard, except where connections were made to standard valves, etc., where the American Standard drilling was maintained. This lessened the cost to the power-plant

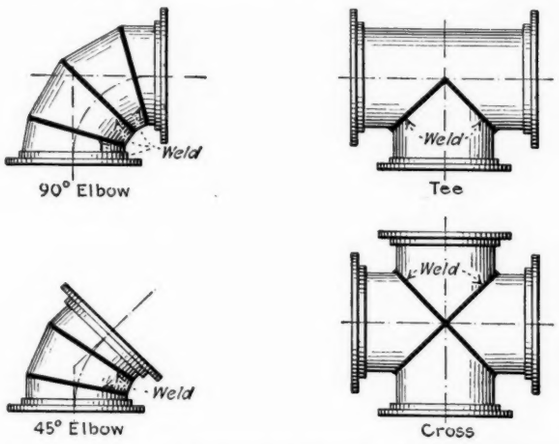


FIG. 9—COMMON FITTINGS BUILT UP BY WELDING

owner. Recently, welding has been substituted for riveting on low-pressure steam fittings. Pipe is either rolled or welded together from steel plate. Outlets are welded onto the pipe. Elbows, offsets and special fittings are made up of welded pipe segments or from welded steel plate.

Fig. 9 shows how the common fittings are made up. The work is all laid out from patterns carried in stock by the fabricators for the standard fittings. Elbows and 45-deg. elbows are cut from straight plate or straight pipe in four or five and three cuts respectively, with no waste. The center-to-face dimensions on any of these fittings can be made either Riveted-Pipe-Manufacturers or American Standard. On any of these fittings small side outlets on tees, crosses or ells, or base ells, etc., present no difficulties.

Complicated Fitting Easily Made by Welding

In Fig. 10, *A* is a special fitting used by the writer and originally laid out for two standard 30-in. 45-deg. elbows, the same dimensions being maintained in the welded fitting. *B* shows how this fitting was cut from 30-in. pipe with four cuts and no waste. About a year after this offset was installed, it was found necessary to put in a 10-in. outlet as shown, which was easily done by burning a hole and welding a nozzle on. Had a cast-iron fitting been used, this would have been impossible without a new fitting.

Three ways of attaching the flanges to the pipe or fitting are shown in Fig. 11, where *A* is a steel flange riveted to the pipe and either calked on edge or calked by welding. In *B* a steel flange is welded to the pipe along two lines as shown. The low-pressure Vanstone flange *C* does away with the difficulty of aligning work in the shop to have holes in opposite flanges in line. It has the same advantage as the similar joint for high-pressure work. These flanges are all made of cast or forged steel, and can be made up to American Standard or to Riveted Pipe Manufacturers Standard drilling. For the American Standard drilling the flange thickness is much less than for cast-iron work. In alteration work, if steel flanges cannot be obtained in time, a satisfactory substitute may be made by curving a steel angle iron to the proper radius and welding the ends together to form a ring.

The thickness of the pipe metal for low-pressure work can be from No. 16 gage to ½ in. thick or over, depending on the diameter and the service, although for steam work for ordinary pipe diameters, ¼-in. thick walls is the general rule, this providing ample metal to resist internal and external corrosion were any to occur.

In a low-pressure installation provision is made for pieces needed in the field to close a pipe run by making a pipe length 6 in. or 12 in. longer in the shop and having one flange on the pipe fastened to it, the other flange being shipped loose. After field measurements are taken, the pipe is burned or cut off to the desired length and the flange is then riveted or welded to it. Another method is to expand one end of the pipe in the shop and, after burning off to the desired length, slip the plain end of another pipe into it and weld.

Tips on Low-Pressure Work

Low-pressure pipe and fittings are, as a general rule, finished in a workmanlike manner. The writer has examined in all about twelve carloads of low-pressure welded pipe and fittings made by different manufacturers, and varying in size from 14 in. to 42 in. in diameter. Most of the material was in as smooth a condition as cast-iron fittings. But in a few instances the weld material was allowed to flow in the pipe and remained without being ground off, causing fins, or the hole was burned an inch or more smaller in diameter than the outlet, either of which causes needless pressure loss.

In making up the joints on low-pressure work with light steel flanges, it is well to use ⅛-in. thick gaskets

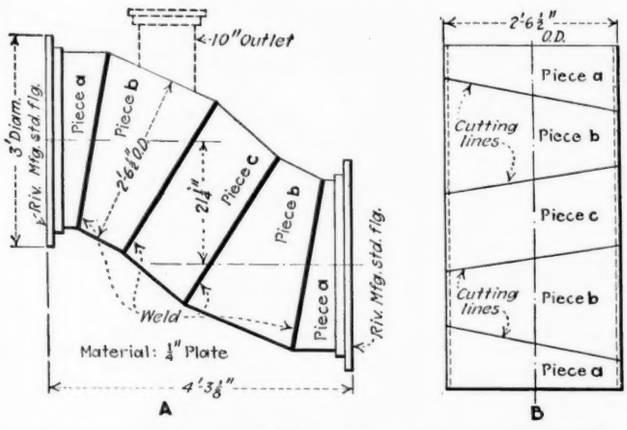


FIG. 10—SPECIAL OFFSET FITTING REPLACES TWO 45-DEG. ELLS

in place of the ⅛-in. ordinarily used, as the joints make up easier with the heavier gasket. All low-pressure work should be subjected to a hydrostatic test pressure of twice the maximum working pressure and made tight against this pressure before using. In making this test, care should be taken (if the pipe is large) that the supports or hangers are not overloaded by the dead weight of the water.

As a rule it does not pay to weld cast-iron pipe or fittings, as certain precautions which make the job expensive, such as preheating and the insertion of steel studs, must be observed to obtain a satisfactory job.

The exceptions are in the case of broken large special castings where an extensive shutdown has to be avoided. In such cases the broken parts are brought together by shrinking links; then a series of small studs are inserted on each side along the crack. The crack is then grooved out, after which a weld is applied, forming a lacing network along the studs and completely sealing the openings.

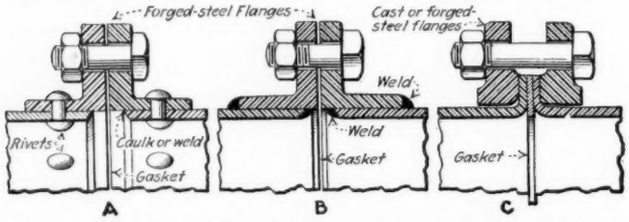


FIG. 11—THREE METHODS OF ATTACHING FLANGES ON LOW-PRESSURE LINES

On alteration work where it is desired to form an opening on a large cast-iron pipe or fitting, this can be done by procuring a flanged cast-iron nozzle with a steel calking strip, or a steel nozzle flanged to the proper radius. Then a hole is cut out by drilling a series of small holes at the point where the nozzle is to be located in the pipe, after which holes are drilled or drilled and tapped to correspond to the nozzle flange and the nozzle is then riveted or patch-bolted to the pipe and calked. Where time is important, a nozzle may be made from a screwed nipple with flange at one end and a steel plate from $\frac{1}{4}$ in. to $\frac{3}{8}$ in. thick curved and welded at the other end, as shown in Fig. 12.

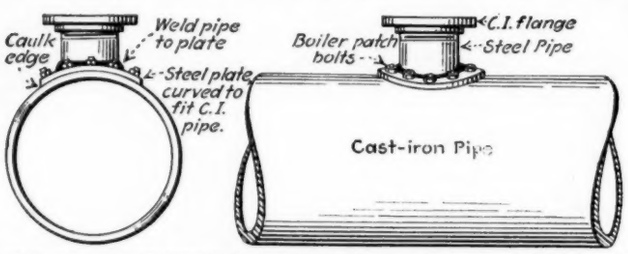


FIG. 12—EMERGENCY METHOD OF FORMING NOZZLE ON CAST-IRON PIPE

In conclusion, it may be said that while the time of the engineer or maintenance superintendent is too valuable to spend on doing the actual welding, he should understand the full possibilities of welding pipe, and it was with this end in view that this article was written. In large industrial plants there are nearly always one or more welders employed on general maintenance work, so that it is an easy matter to have pipe welded in the power plant. In an ice plant or the power plant of a public utility, the engineer can generally call on a reliable welding concern in his own community to do his work, provided he supervises the operations.

Drying and Purifying Machines for Insulating and Other Oils

The illustrations herewith show a machine recently developed by the Westinghouse Electric & Manufacturing Co. for dehydrating and filtering insulating oil, such as used in transformers, oil circuit breakers, regulators, etc.

The machines are made in five standard sizes, divided into two classes according to the size of the filter paper. The three larger sizes are rated at 10, 20 and 30 gal. per min. and the smaller sizes at $2\frac{1}{2}$ and 5 gal. per min.

The filter press proper is made up of a series of flat cast-iron plates and frames assembled alternately with filter papers between them. By means of a screw and lever and a movable cast-iron end block, the plates, frames and papers are forced tightly together. The plates and frames are cast with holes *A* and *B* in two of the corners. When the plates and frames, with the filter papers between them, are assembled, the holes *A* and *B* form respectively the inlet and outlet of the oil.

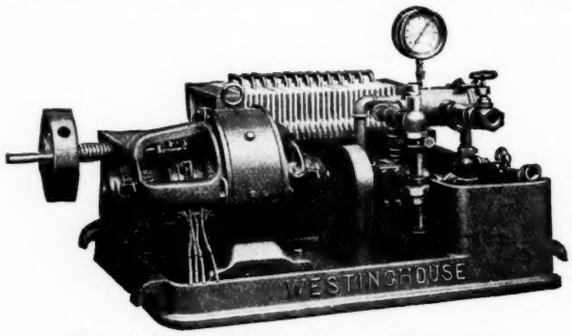


FIG. 1—TEN-GALLON OIL DRYING AND PURIFYING OUTFIT

The oil entering under pressure at the top corner through the inlet formed by the holes *A* in the frames, plates and filter papers, enters by means of the *a* openings in the frames and fills each of the chambers formed between each set of two filter packs. From these chambers there is no outlet opening and the oil is consequently forced through the filter papers that form the sides of the chambers. The filter paper thus takes up the moisture and screens out the sediment from the oil. After passing through the filter paper, the oil flows along the grooves formed by small pyramids constructed on both surfaces of the plates to the opening *b* in the lower corner of the plates and thence out through the outlet *B*.

FIG. 2—ASSEMBLY OF FILTER-PRESS PLATE AND FRAME, SHOWING THE FORMATION OF OIL CHAMBERS BETWEEN EACH TWO FILTER PAPERS

The filter paper used is a special grade of white blotting paper about 0.025 in. thick. Five sheets, cut to the proper size with holes punched to correspond with the holes in the filter plates and frames, are used between each plate and the adjacent frame.

This equipment is adapted to the treatment of insulating varnishes and compounds that have a sufficiently low viscosity, also lubricating oil, petroleum and other liquids.

Determining the Economical Interval Between Cleanings of Condenser Tubes

By C. E. COLBORN
Testing Operator, Efficiency Department, West Penn Power Company

Sediment deposited on condenser tubes by cooling water is frequently a matter of continual daily accumulation. The best vacuum obtainable then steadily decreases. The steam consumption of the prime mover therefore is gradually increased, representing a loss in dollars, which is balanced against the cost of cleaning tubes. The most economical period between tube cleanings depends upon how fast the sediment is deposited, the cost of the vacuum loss which varies with the relative load to a large extent, and the cost of cleaning tubes. Economical operation usually requires a seasonal variation in cleaning schedule.

CONDENSER tubes usually foul more frequently as a rule in the summer than in the winter and, unfortunately, it is the former time of the year that it is most desirable to maintain a high rate of heat transfer on account of warmer circulating water. In many plants the turbines and engines are not designed to utilize the high vacuums that can be obtained during the winter months when the temperature of the circulating water entering the condenser is below 40 deg. F. During these months the condensers

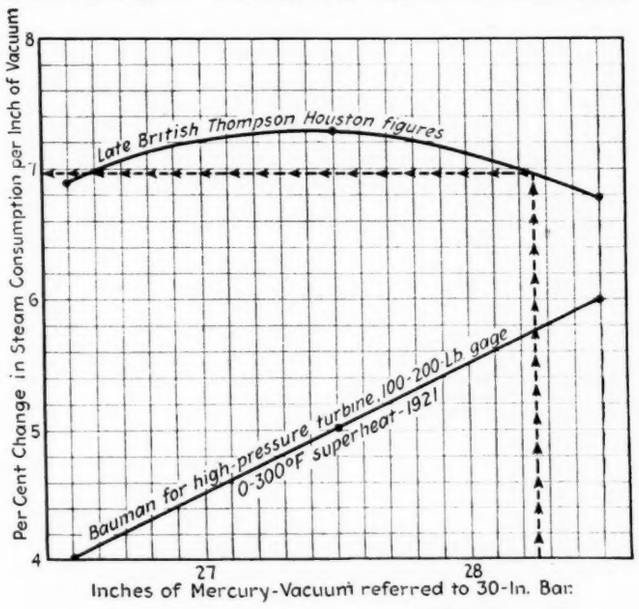


FIG. 1—EFFECT OF VACUUM CHANGES ON STEAM CONSUMPTION, AT CONSTANT INITIAL STEAM CONDITIONS

should be cleaned at comparatively long intervals if at all, for real economy. A cleaning schedule that does not make some provision for varying the frequency of cleaning throughout the year will not result in the most satisfactory and economical operation of the unit.

In determining the most economical frequency of cleaning a condenser, it is necessary to know three things: First, the cost of cleaning; second, the daily cost of the loss due to operating with a given reduction

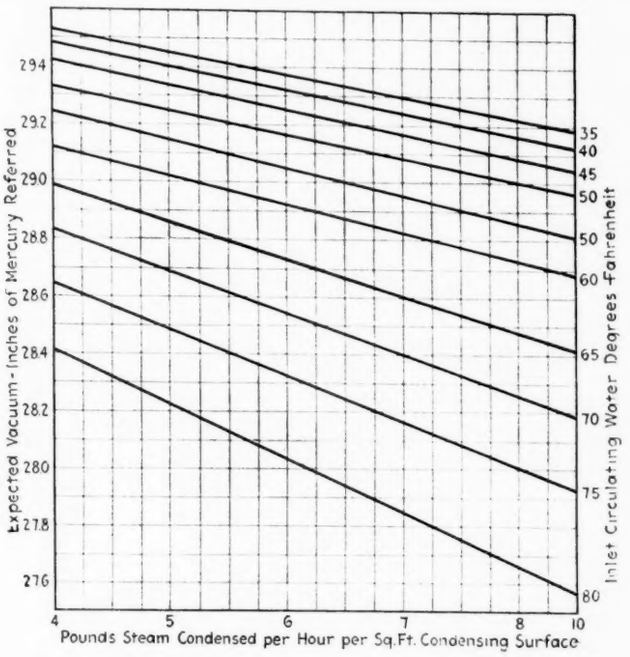


FIG. 2—EXPECTED VACUUM OF A LARGE SURFACE CONDENSER, RESULTING FROM VARIOUS COOLING-WATER TEMPERATURES

in vacuum; third, the daily average reduction in vacuum due to the fouling of the tubes.

In stations having a system of cost accounting, it should be very easy to ascertain the cost of cleaning the condenser, the first necessary item. If it is necessary that the turbine be shut down especially for cleaning the condenser, the cost of taking the unit out of service should be added to the labor costs to obtain the total cost of cleaning.

How the Economical Vacuum Is Limited

It is well known that increasing the vacuum also increases the available heat head in the turbine, since it lowers the exhaust temperature and pressure, allowing the steam to give up more energy. This also increases the volume of the exhaust steam, the specific volume being almost twice as great at 29 in. vacuum as at 28 in. The velocity of the steam leaving the last row of blades varies directly with the volume of the exhaust steam. The kinetic energy of this steam leaving the last row of blades is proportional to the square of the velocity and is almost entirely lost in shock and

eddies in the condenser and turbine exhaust chamber. This loss is small when the velocity of the leaving steam is small, but increases rapidly with an increase in vacuum until a point is reached where the increase of this loss, commonly called the leaving loss, entirely offsets the gain in available heat head due to any further increase in vacuum. This point is sometimes called the limiting vacuum.

The velocity of the steam leaving the last row of turbine blades is also proportional to the load on the turbine, and thus the lost kinetic energy in the steam will be less for low loads than for higher loads. Owing to this fact the limiting vacuum is different at each load, being lower at higher loads. These general rules also apply to the reciprocating engine. The limiting vacuum, as well as the corrections at vacuums below the limiting vacuum, will vary greatly with different designs of turbines and engines.

Suppose, for example, the turbine is designed to operate at its most economical load at 29 in. vacuum, while the load carried in operation averages twice the most economical load. The turbine then will require approximately twice as much steam—actually, somewhat more than this, since it will operate at less efficiency. The limiting vacuum will then lie somewhere near 28 in. Since steam at 28 in. has approximately half as much volume as at 29, twice the weight of steam at 28 in. would pass the last row of buckets at about the same velocity as the normal amount at 29 in. Turbines ordinarily operate close to the rated most economical load. Otherwise, curves of limiting vacuum should be obtained from the manufacturer, together with correction factors for figuring the loss due to vacuums lower than the limiting vacuum.

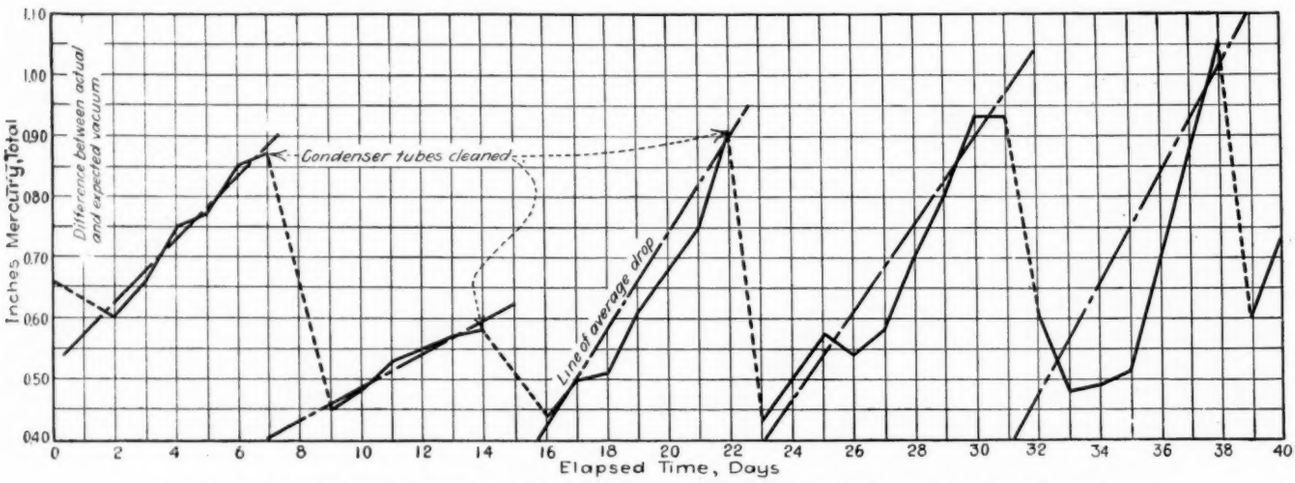


FIG. 3—AVERAGE DAILY DECREASE IN VACUUM SHOWN BY PLOTTING DAILY DIFFERENCE BETWEEN EXPECTED AND ACTUAL VACUUM

Figuring Loss Due to Reduced Vacuum

Fig. 1 gives two vacuum-loss correction curves which are generally applicable to large turbines. One is according to Bauman and the other is calculated and plotted from late figures of the British Thomson-Houston Co. Curves similar to these can be furnished by the turbine or engine manufacturer. From these curves one can easily calculate the cost of daily operation of the machine at a one-tenth-inch reduction in the vacuum, the second item. If the turbine load does not vary over too wide a range, it will be sufficiently accurate in obtaining this cost to find the vacuum correction factor corresponding to the average daily load and vacuum and multiply it by the number of daily service hours of the turbine or engine. This factor multiplied by the cost of generated electrical power will give the daily cost of operation under the reduced vacuum.

In obtaining the daily reduction in vacuum due to the fouling of tubes, it is desirable to correct the actual vacuum for changes in the circulating-water temperature. This can best be done by using curves similar to those shown in Fig. 2, copies of which for any condenser can be furnished by its manufacturer. These curves show the vacuum that should be obtained with different amounts of steam supplied to the condenser, and with inlet circulating-water temperatures as indicated, and apply to a large radial-flow two-pass surface condenser operating with a water velocity of about 5.5 ft. per sec. through the tubes. The difference between the actual and expected vacuums should be plotted against time on cross-section paper. Fig. 3 shows such a group of curves obtained from a large condenser similar to that previously mentioned. The average daily drop in vacuum can be obtained by drawing a straight line through the curve as shown in the figure.

Assume the following symbols:

C represents the total cost of cleaning a condenser.

L represents the daily cost in dollars of operating the prime mover at a one-tenth-inch reduced vacuum.

R represents the average daily reduction in vacuum, expressed in inches of mercury.

X represents the most economical period in days between cleanings, which is to be found.

The sum of the cleaning cost per day of operation, and the cost to the plant per day of operating at a low vacuum, is $\frac{C}{X} + \frac{RLX}{2}$.

The object of the calculation is to find when this cost is least. Cleaning too often will make the sum of these costs excessive. The efficiency will be reduced little or practically not at all by a small accumulation of scale which collects slowly and the cost of frequent cleaning will be excessive. On the other hand, cleaning intervals that are too long will cause excessive loss due to decreased turbine efficiency, especially with rapid scale formation. The correct time interval will make the sum of these costs under any conditions a minimum.

All quantities can be determined except X. This is a variable, while the others are constant for any given

condition. It is assumed that the daily accumulation of scale is uniform which will be found very nearly true. By using operations of differential calculus, it can be proved that $\frac{C}{X} + \frac{RLX}{2}$ is least when $X = \sqrt{\frac{2C}{LR}}$. From the known quantities, C, L and R, we may then readily determine X, the most economical cleaning period, in days.

This equation is based on the assumption that the daily drop in vacuum is constant, an assumption which will be found to very nearly fall true, especially in cases where the fouling of the tubes is bad, as has been borne out by the writer's experience. Variations in the amount of air leakage and errors in taking the condenser readings will cause the slope of the curve of the difference between the actual and expected vacuums to vary from day to day, but will not greatly affect the average slope obtained for a number of days. The readings should be taken at nearly the same load as far as possible at all times, since only by using a constant base can the results of daily tests when plotted be made to give a real picture of what is going on in the condenser.

Assuming that for a 15,000-kw. unit a 20,000-sq.ft. condenser had been supplied, and that while in operation with a 12.5-lb. water rate the condenser receives 160,000 lb. of steam per hour at a vacuum of 28.25 in. referred to a 30-in. barometer. A vacuum increase between cleaning periods was obtained as shown in Fig. 3, days 2 to 7 inclusive. Over this period the average daily increase was 0.057 in., so that $R = 0.057$ inch.

At the vacuum obtained, according to the British Thomson-Houston curve of Fig. 1, the increase in steam consumption is 6.95 per cent per inch reduction in the vacuum, or 0.695 per cent per one-tenth inch.

The daily total steam consumption would be 160,000 $\times$ 24 = 3,840,000 lb. Therefore the daily increase in steam consumption is 0.00695 $\times$ 3,840,000 = 26,688 lb. per one-tenth inch reduction in vacuum.

If 1 kw.-hr. generated costs \$0.0066 and our turbine water rate is 12.5 lb., our steam cost will be \$0.0066 $\div$ 12.5 = \$0.000528 per pound delivered at the turbine. Hence the increased daily operating cost per one-tenth-inch reduction in vacuum is 26,688 $\times$ \$0.000528 = \$14.09, which is L.

We have now found the necessary items for our calculations: L equals \$14.09; R, 0.057, and C, \$8.

$$X = \sqrt{\frac{2 \times 8}{14.09 \times 0.057}} = 4.47 \text{ days.}$$

This indicates that it is advisable to clean the condenser every five days.

Now for some reason during the next period the drop in vacuum was as shown in Fig. 3, days 7 to 14. During this time R was equal to 0.032, and with L and C the same as before

$$X = \sqrt{\frac{2 \times 8}{14.09 \times 0.032}} = 5.97 \text{ days,}$$ which would show it best to clean the condenser every six days.

Owing to the number of variable factors that enter into the calculation of the most economical period between cleanings, it will be found advisable to make frequent determinations of its value.

The curves in Fig. 4 will greatly simplify the calculation of the economical frequencies of cleaning. Begin with R at the left, 0.057, follow arrows to the right to the curve L of 14.09, which gives $LR = 0.804$. Drop vertically down to C curve of \$8 and read at the right 4.5 days as the theoretical period for economical cleaning of condenser tubes.

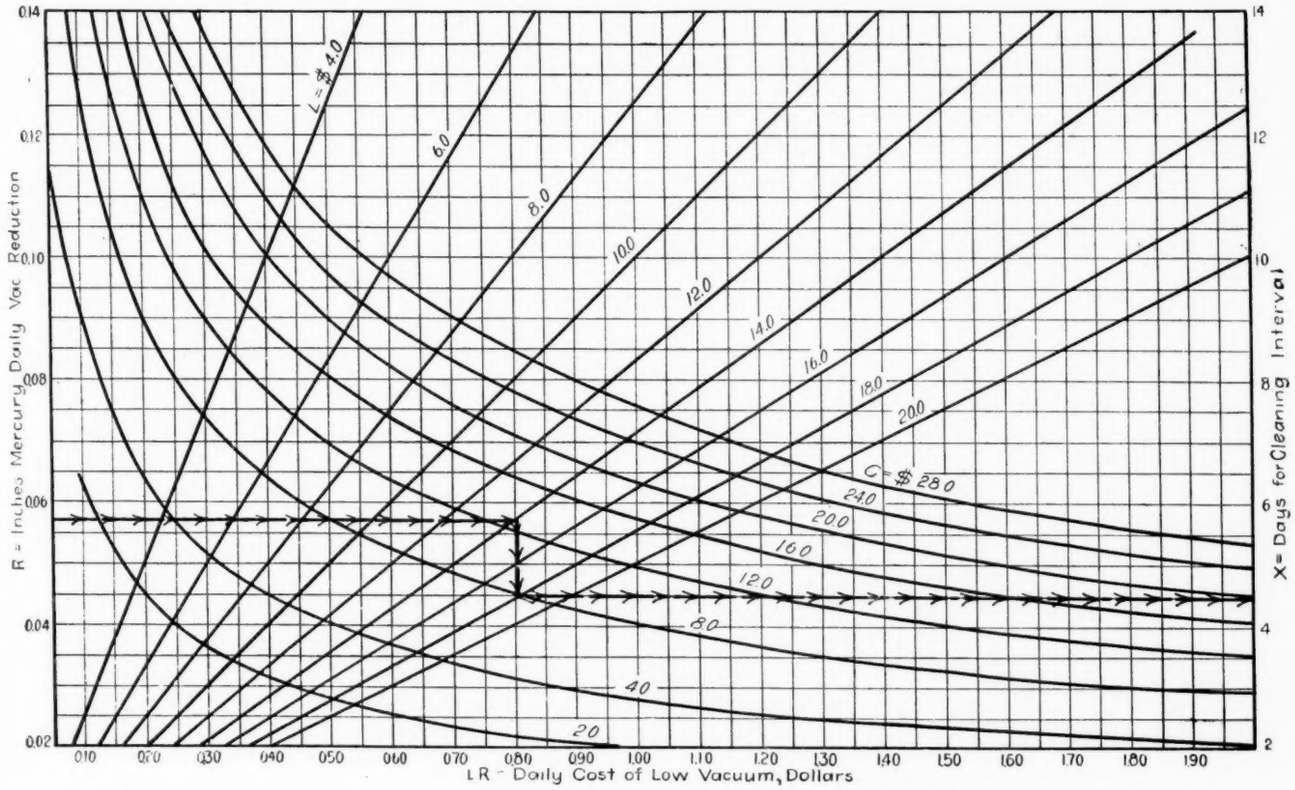


FIG. 4—ECONOMICAL CLEANING PERIOD QUICKLY OBTAINED FROM KNOWN VALUES OF R, L AND C

Parallel Operation of Generator with Purchased Power

Why the Meters in a Power Plant Operating in Parallel with a Transmission Line Gave Inconsistent Readings

By VICTOR H. TODD
Engineer, Westinghouse Electric & Manufacturing Company

WHEN a power load is supplied from a parallel-operated transmission line and turbo-generator, the load may be readily transferred from one to the other by manipulation of the generator's speed. Many such systems are a gradual growth rather than a unit design, and the instruments may be connected in such a manner that the transfer of load will give surprising indications unless it be clearly understood what the instruments indicate and whether they are actuated by the total load, the portion supplied by the transmission line or the portion supplied by the generator.

Such a case is given by the figure which shows the schematic layout of a generating station. There is a 1,500-kva. 2,300-volt turbo-generator at *A*, whose output passes through a transformer bank, which steps up the voltage to 13,200 and feeds into the north, east and south transmission lines. These lines supply a mixed load of synchronous motor-generator units, induction motors and lamps, which are located in mines some distance away. There is also about 500-kw. local load taken from the two-phase bus as indicated. Also, supplying these transmission-line loads is a 33,000-volt incoming transmission line fed from a station about 30 miles distant. This power is stepped down to 13,200 volts through a bank of transformers and tied into the 13,200-volt bus by the oil switch *G*. All synchronizing and protective connections have been omitted for the sake of clarity. The alternator is equipped with a watt-hour meter *F* and a power-factor meter *H*, while the incoming transmission line has a watt-hour meter *E*, an indicating wattmeter *D* and an ammeter *C*.

The engineer in charge of this equipment observed the following conditions and asked for an explanation of the apparently erratic behavior of the apparatus:

"Case 1: When the alternator and transmission line are operating in parallel and supplying a total load of 1,500 to 2,500 kva., divided between the station and the transmission line, all meters check out correctly and the indicated power factor at *H* is 0.80 to 0.85. By slightly increasing the turbine's speed and reducing the excitation, the power factor can be raised to unity, the indicated amperes on the generator decreased and the integrating wattmeter *F* increased slightly in speed, while the ammeter at *C* increased and the reading of indicating wattmeter *D* decreased.

"Case 2: Assuming a load of 1,200 kilowatts divided between the turbine and transmission line. If the turbine is speeded up and excitation is reduced until all load is transferred to the turbine, the wattmeter at *D* will read zero, the integrating wattmeter *E* will stop and the power factor will be unity. The ammeter at *C* will, however, indicate from 20 to 26 amperes, and if under such conditions the oil switch at *G* is opened, the power factor will drop to 0.80 or less with increased amperage on the turbine.

"In case 1 when the power-factor meter indicates unity, can this be taken as the system power factor or does it apply to the turbine only? In case 2 is the cur-

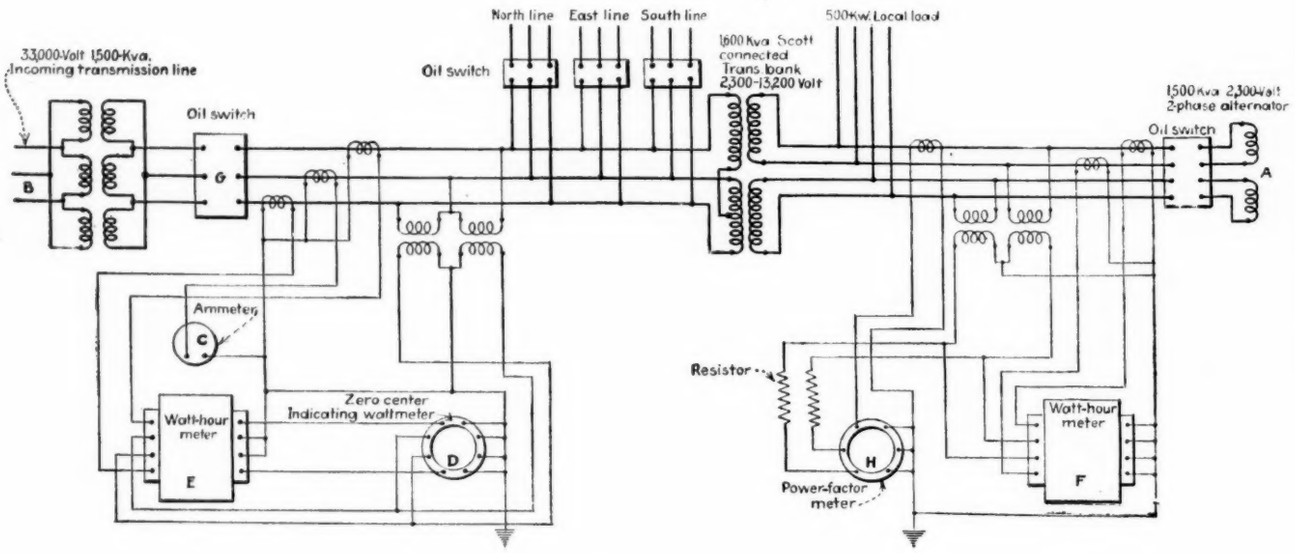


DIAGRAM OF CONNECTIONS OF ALTERNATOR, TRANSMISSION LINE AND LOADS

rent indicated by ammeter *C* synchronizing or magnetizing current, or is this due to wattless or circulating current set up between the transmission line, transformers and turbine?

"When oil switch *G* is opened and the power factor drops from unity to 0.80 or less, why does this increase the load on the turbine? Some of my colleagues differ with me on this point, claiming that it does not, but only increases the kva. load and affects the generator capacity only. However, the action of the steam-flow meter and turbine governor proves otherwise. Under this condition would it be correct to assume that the transmission line was supplying the required wattless or magnetizing current necessary for the system in order to maintain unity power factor on the turbine? In order to determine just how much energy was being supplied by the transmission line under such conditions, what kind of metering equipment would be recommended?"

Explanation

In making an analysis of this problem, the first step is to consider just what the instruments will indicate and to determine what portion of the load actuates them. Consider first the power-factor meter *H;* the way this is connected, it will indicate only the power factor under which the turbo-generator is operating. When part of the load is supplied by the incoming transmission line, then the power factor meter does not give the least idea of what the power factor of the load is, nor of the power supplied by the transmission line. However, when the switch *G* is opened, then the turbo-generator supplies the entire load and the power-factor meter indicates the power factor of both turbo-generator and load. From this it is evident that the power factor of the load is about 80 per cent.

The next point to remember is that the power factor of the load will be changed but slightly by a manipulation of the supply. Therefore, if the load is supplied by both turbine and transmission line, the power factor of the load is 80 per cent regardless of the power-factor indication on the turbine. Now, if the 80 per cent power-factor load is supplied partly by the turbine working at 80 per cent power factor and by the transmission line, it follows that the portion of the load drawn from the transmission line will also be at 80 per cent power-factor. This is the assumption upon which Case 1 is based.

Proceeding further, it is proposed to speed up the alternator. This, of course, cannot be done and still stay in synchronism, without increasing the speed of all alternators on the system, as the frequency is a function of the speed and therefore both transmission line and alternator must be at absolutely the same frequency to stay in synchronism. What can be done, however, is to supply more steam and decrease the excitation until the power factor on the turbine is unity, or 100 per cent. This, of course, will decrease the turbine amperes and cause it to take more kilowatt load, thus resulting in a decrease in the kilowatt load supplied by the transmission line. The power factor of the load must still be 80 per cent, therefore, if the portion supplied by the alternator has a 100 per cent power factor, it must combine with a lower power-factor load from the line to produce the 80 per cent power-factor load on the system. This lower power factor produces an increased current on meter *C*. The value of this power factor depends on the weighted magnitudes of the portions of the load supplied from the alternator at 100 per cent and the portion supplied from the transmission line at a lower power factor. If these portions were equal, then the part supplied by the transmission line would be at 60 per cent power factor.

In case 2, if the power factor of the alternator is unity and the actual load supplied is 80 per cent, then the reactive current demanded by the 80 per cent power-factor load must be supplied from some other source. This source is the transmission line; and since the line is not supplying any kilowatts, as ascertained by zero reading of *D* and *E*, then the reading of 20 to 26 amperes on *C* represents the reactive current demanded by the 80 per cent power-factor load, circulating between transmission line and load and which is refused by the turbo-generator operating at 100 per cent power factor. There is no wattless or circulating current in the turbine alternator when the power-factor meter indicates unity.

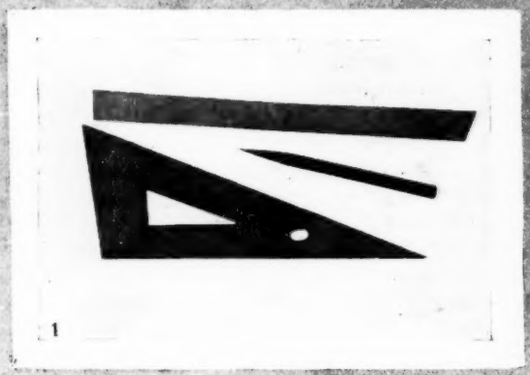
Without voltmeter readings it is not just clear what takes place when switch *G* is opened, but unless the field rheostat is manipulated, apparently the following happens; First, a heavier current flows in the turbo-alternator, due to the lower power factor, and this to a degree demagnetizes the fields, resulting in a lower voltage. The horsepower load remaining the same, greater current is demanded at this low voltage, thus again resulting in greater I^2R losses in the line, transformers and generator. This may or may not change the power factor of the load, depending on the characteristics of the motors; that is, the proportion of synchronous and induction motors, and other load.

Regarding the indications of the other meters, meter *F* registers the total energy supplied by the turbo-alternator, meter *E* registers the kilowatt-hours supplied from the transmission line, while meter *D* indicates this power. Meter *C* shows the current in one phase of the incoming transmission line, whether it be at 100 per cent or 0 per cent power factor, and its indications may be used as a basis of figuring transformer and line losses if desired. It would be interesting to connect in a power-factor indicator with *E* and *D* to determine the power factor of the load drawn from the line, or a wattmeter or watt-hour meter at this point to determine the reactive kilovolt-amperes or the reactive kilovolt-amperes hours to see how they compare with the kilowatts or kilowatt-hours.

Many engineers do not realize the magnitude of the kilowatt losses at low power factors, but the following is typical: The wattmeters showed that a certain plant was carrying a load of 11,000 kw., at a power factor of 70 per cent, including all generator, transformer and line losses. This load was raised to 100 per cent and the kilowatts dropped to 9,800, or a saving of 1,200. This amounts to real money and plainly shows why a turbine takes more steam at low power factors to supply a given kilowatt load.

Four combined hotwell and boiler-feed pumps for the Trenton Channel Plant of the Detroit Edison Co. are being built by the De Laval Steam Turbine Co. Two will be motor- and two turbine-driven. Hotwell pumps at the outboard bearings rated at 1,500 gal. per min. at 180 ft. head force the condensate through heaters where the temperature is raised by extracted steam. The six-stage 8-in. single-suction boiler-feed pumps, directly coupled and in separate casings, deliver 1,300 gal. per min. at 1,090 ft. head.

MAKING A STRAIGHT-EDGE FOR READING NOMOGRAMS



1

2

1

The first step is to get a piece of thin celluloid (*old photo film washed free from gelatin will do*) and cut it 1-inch wide and about 12-in. long. Then mount a steel needle point in a wooden handle or mechanical pencil.

2

Using the needle point and a straight-edge, scratch a line down middle of film with short cross scratches at center and ends.

3

Scrape off the burr with razor blade or a sharp knife.

3

4

Smear drawing ink over the scratch and let it dry.

5

Rub off excess ink with a clean rag, leaving sharp black line on film. Make fine hole with needle point at center and end cross scratches and mark top surface of film, (*surface opposite the scratch*). This completes construction.

6

Use strip, smooth side up, to read any collinear diagram, pivoting black line about needle placed at given point on scale.

4

5

6

Straight-Edge for Reading Charts

The pictures and captions on the opposite page explain the construction of a convenient straight-edge for use in reading charts. While the directions for the actual construction of the straight-edge are perfectly simple and straightforward, it is possible that some readers may be unfamiliar with term "Nomograms" in the heading, and the term "Collinear Diagram" in the caption of Fig. 6. Many years ago a French mathematician by the name of d'Ocagne made an extensive study of the solution of mathematical problems by means of diagrams. He called the subject "nomographie," from which the term "nomogram" has been derived. In plain English, a nomogram is any kind of a diagram used to solve a given formula. It is what is often called a "chart." The collinear diagram is one type of nomogram, consisting of several lines with scales on them, the solution being obtained by stretching a straight-edge from one scale to another. This is the type of nomogram for which the straight-edge shown on the opposite page is particularly adapted. A good example of a collinear diagram is the English-Metric Conversion Chart published in the Sept. 4 issue of *Power*. A few reprints of this on bristol board are still available for free distribution to subscribers on request.

New Republic Gages To Indicate and Record Draft

New principles of draft measurement have been incorporated in two instruments recently brought out by the Republic Flow Meters Co., of Chicago. One of these indicates the draft on a 12-in. dial; the other, which operates on the same principles as the indicator, records the draft on a circular chart. These instruments are shown in Figs. 1 and 2, respectively.

The indicator involves the principles of the bell float, the balance and the plumb weight, so combined as to indicate the draft in the furnace independent of the height of the liquid forming the seal. This is accomplished by balancing the buoyant effect of the float and providing a weight acting on the beam of the balance to resist the acting force of the pressure difference between the furnace and atmosphere.

Reference to Fig. 4 will show diagrammatically how the gage operates. Here *A* represents a bell float suspended from one end of the beams *BC* and extending into the oil chamber *D*. Leading up into the interior of the float is a small tube *E* connecting with the source of pressure to be measured. To the opposite end of the beam is attached the member *F*, which, having the same cross-section of material as the walls of the float, displaces an equivalent volume of liquid, so that the weights of the two elements *A* and *F* remain practically the same for any change in the oil level. As the float and the counterweight move in opposing directions, there will be a small difference in buoyancy proportional to the motion, but it is claimed that the effect is included in the value of the weight *G*, which is attached rigidly to the beam at the fulcrum *H*. Under these conditions the oil in the chamber may be depleted fully one-half inch from the original level without affecting the operation or accuracy of the instrument. It will be noticed also that a movable weight *I* is extended from the beam *BC* for adjusting accurately the weight of the member *F* with that of the float, so that the weight *G* offers the main resistance to the movement of the beam.

Both Figs. 1 and 4 show the float in its maximum upward position, this being obtained with atmospheric pressure within the float and the pointer at zero on the dial. As the pressure is reduced within the float, atmospheric pressure on the outside tends to push it downward into the oil against the counterbalancing action of the weight *G*. As the rod carrying the weight is rigidly attached to the beam, it passes through an angular movement equal to that of the beam, carrying the weight off center, so that the movement of the dial hand is a function of the relation between the position of the beam and the position of the counteracting weight. The angular relation between beam and bar is selected with care, so that the dial hand will make practically equal steps for equal increments of pressure difference. The gage responds immediately to a pressure variation, the pointer coming to the new position with no noticeable hunting.

The instrument shown in Fig. 1 was designed to measure a negative pressure of 2 in. of water indicated over an arc of 288 deg. A gear ratio of 1 to 5 in the mechanism determines the motion of the beam as one-fifth of this travel, or 57.6 deg. In the recorder the mechanism is the same, with the exception that the gear is omitted. The recording arm is attached directly to the beam and describes the same arc of 57.6 deg.

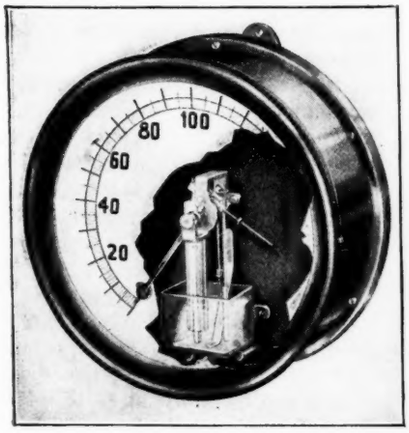


FIG. 1—REPUBLIC DRAFT INDICATOR

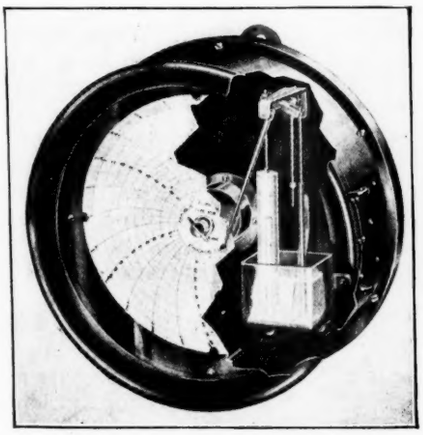


FIG. 2—PHANTOM VIEW OF DRAFT RECORDER

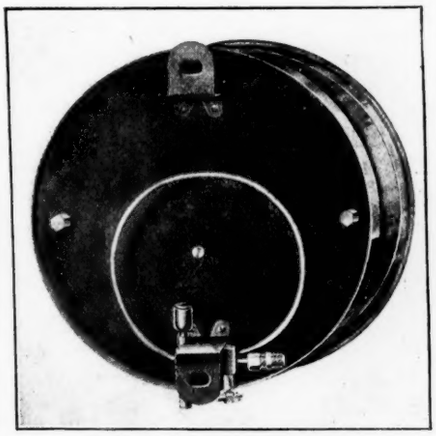


FIG. 3—BACK VIEW OF INSTRUMENT

over the chart which by a clock movement is revolved once in 24 hours.

When placing the instrument in operation, the chamber shown in Fig. 1 is filled with oil to the height of the inlet and a thumbnut at the back of the case manipulated to release a locking device consisting of a false bottom in the oil chamber, which is pressed upward

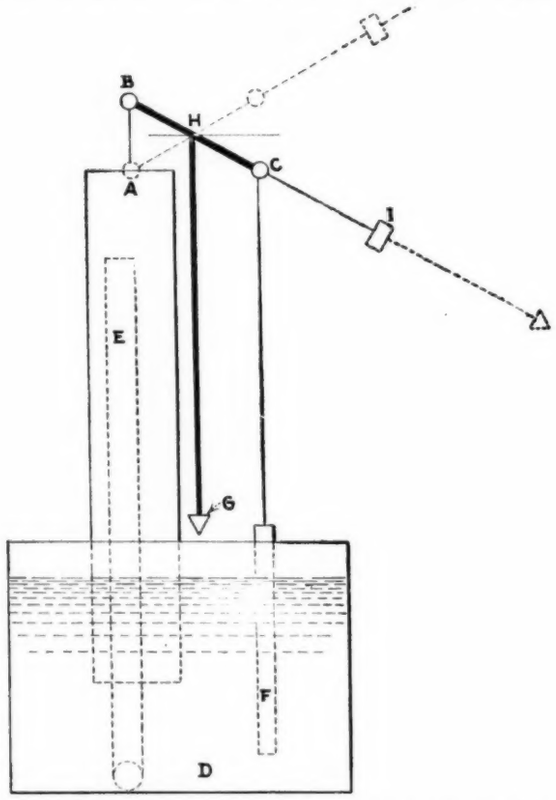


FIG. 4—DIAGRAM SHOWING PRINCIPLES OF OPERATION

against the float to keep the beam in a horizontal position during shipment. Reference to Fig. 3 will show at the back of the case a flexible coil to which the furnace line is attached. This provision allows the instrument to be shifted into place without disturbing the piping.

For subsequent calibration a double valve is provided at the inlet, which simultaneously shuts off the draft and opens the float to atmospheric pressure. When the instrument is in a true vertical position, the hand will point at zero and no other leveling is necessary.

Four models of the gage are under production—three instruments with dial ranges from 0 to 1 in., 0 to 2 in. or from —1 to +1 in. of water and a differential instrument employing two floats, the second float taking the place of the balancing member previously mentioned, but with no other change in the mechanism.

The relation between the pressure, volume, weight, temperature and molecular weight of gases is given by the following formula:

$$P \times V = \frac{10.7 \times W \times T}{m.\ w.}$$

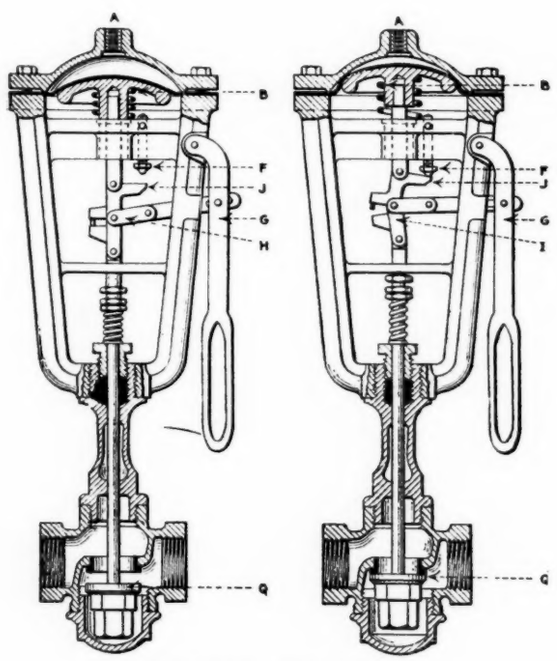
where P = absolute pressure, lb. per sq.in.;
V = volume, cubic feet;
W = weight, lb.;
T = absolute temperature, deg. F. + 460;
$m.\ w.$ = molecular weight.

Oil-Burner Shutoff Valve

In low-pressure heating plants where the fireman is compelled by reason of other duties to be absent from the boiler room, there is a danger of an explosion where oil is the fuel. It is impossible, on light loads and at low steam pressures, to so regulate the oil-burner valves that all danger of the fire going out is absent. If the flame does disappear and the oil continues to flow, the vapor accumulating in the heated flame will explode.

The C. J. Tagliabue Manufacturing Co., Brooklyn, N. Y., has brought out the automatic shutoff valve shown in the illustration. This valve is placed in the oil line to the burner, while the space above the diaphragm *B* is connected through *A* to the steam line. As shown at the left, the valve is in operating position with the oil-line valve *Q* open by reason of the steam pressure above the diaphragm forcing the valve stem downward.

In case the steam pressure falls, the diaphragm *B* is forced upward by its spring, carrying the valve stem with it. The stem is in two parts connected by hinge



OIL BURNER SHUTOFF VALVE

H. When the stem rises enough to permit *J* to strike the pin *F*, the hinge opens to the position shown at the right. If steam pressure is exerted upon the diaphragm, the upper section of the stem will move downward, but without altering the position of the lower part of the stem until the hand lever *G* pulls the hinge into the position shown at the left. The valve, as a consequence, shuts automatically upon a drop in steam pressure, but must be replaced in the open position by manual operation.

William Arnett, in a recent issue of the *American Machinist*, states that plate glass, spring steel and other hard materials can be successfully drilled if the drill is ground to the proper angle and liberally supplied with a mixture made by dissolving a quarter's worth of camphor gum in a half pint of alcohol, then adding one-half pint of lard oil.

The Meaning of Atmospheric Pressure

By T. M. GUNN

THE most common of all pressures, and at the same time the pressure that is the least observed by our senses, is that of the atmosphere. We are not conscious of it because we are so completely accustomed to it.

Although the exterior surfaces of the human form are subjected to a total load—that is, the product of area by pressure—generally exceeding thirty thousand pounds, the natural laws that govern fluid pressure result in such a perfect balance that this tremendous load does not move us an inch one way or another. Instead of being crushed down by this burden, we are actually buoyed up, like a balloon, to the extent of about three ounces.

It is well for engineers to be familiar with the laws of atmospheric pressure. Every time a pressure or vacuum gage is used, the engineer learns only the difference between the pressure or vacuum in question and atmospheric pressure, the latter being the starting point for such measurements. If the latter pressure is not known, the observed gage readings cannot be reduced to true, or absolute, pressures.

It is often assumed that the pressure of the atmosphere is 14.7 lb. per sq.in. (29.9 in. of mercury). This is the average or mean pressure at sea level, but from time to time the atmospheric pressure at such a location changes with weather conditions, such as temperature, humidity and winds.

Between one location and another the local mean atmospheric pressure varies, dependent on the latitude and altitude of the place and on the mean temperature and humidity conditions.

The pressure of the atmosphere is, like the pressure in a body of water, merely the result of the weight of a column of air extending upward to the limits of the atmosphere. Unlike water, however, air is highly elastic and, owing to the decreased pressure, is less dense at greater altitudes.

If we rise to a level of about 3½ miles (18,750 ft.) above sea level, the atmospheric pressure is reduced to one-half. In other words, at that altitude the weight of air below us is equal to that above. If we could rise another 3½ miles the pressure would again be cut in half; that is, the pressure would be one-fourth that at sea level, three-fourths of the air lying below our level. At a level 3½ miles higher, or a total of 10½ miles above sea level, the pressure is only one-eighth; and at 14 miles it is only one-sixteenth part of that at sea level, or less than one pound per square inch.

For those who like to see the formula, here it is:

$$h = 62{,}580 \log \frac{29.9}{B}$$

In this formula, h = altitude above sea level in ft. and B = atmospheric pressure, inches of mercury. To use the formula with pressure expressed in pounds per square inch, use 14.7 instead of 29.9 for the numerator of the fraction. This formula is based on the assumption of a mean atmospheric temperature of 50 deg. F.

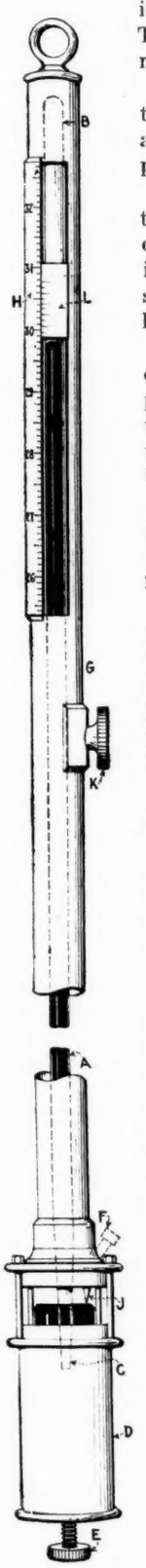


FIG. 1 MERCURIAL BAROMETER

The chart, Fig. 2, gives the same results as this formula, with the addition that it takes account of the mean temperature of the atmosphere.

To show the use of the chart, Fig. 2, assume that in a climate where the mean temperature of the atmosphere is 40 deg. F., a power plant is to be installed at an altitude of 5,500 ft. above sea level. What is the probable mean barometric height?

The solution is as follows: On the top scale of the chart, Fig 2, find the point *A* for a temperature of 40 deg. F. and on the bottom scale the point *B* for an altitude of 5,500 ft. Draw the line *A-B*, and read at the intersection *C* the barometric height, 24.32 inches.

To reduce barometric height to pounds per square inch, multiply by 0.4906, which is the weight of one cubic inch of mercury at 32 deg. F. $24.32 \times 0.4906 = 11.93$ lb. per square inch.

If, when using this chart, the mean atmospheric temperature is not known, it is customary to assume a temperature of 50 deg. F., which is marked with the letter *D*. Using the point *D* instead of some other value such as 40 deg. F., makes the chart equivalent to the formula that has been given.

The instrument by which the pressure of the atmosphere is measured is the barometer. Of this class of instruments there are many types, generally classified as mercurial and aneroid. An aneroid barometer, in principle, is like a pressure gage of the diaphragm type, but with the air within the gage thoroughly exhausted to a vacuum and the connection closed. With practically zero pressure within the diaphragm and atmospheric pressure outside, deflections of the needle indicate variations in the pressure of the atmosphere. Aneroid barometers are not altogether reliable and may have unknown variations as the temperature changes, but they are very convenient and many of them are accurate enough for practical use when the corrections are small or of known value.

The most reliable type of barometer is that using mercury. Although the mercurial barometer must be handled with care and must be corrected for temperature, etc., it is not difficult to learn when the instrument is out of order, and the corrections can easily be made to a high degree of accuracy. Such instruments are becoming common in large power plants where accurate measurements are essential and should be more generally used in keeping daily and special test records.

The principle of the mercurial barometer, without going into the details of construction, which vary widely with different makers, is illustrated in Fig. 1.

The essential parts are as follows: A vertical

glass tube *A*, somewhat over 30 in. long, has its upper end *B* closed and the lower end *C* open and dipping in the mercury, held in the cistern *D*, which has walls partly of glass, and is provided with some means, such as a tight piston or flexible diaphragm and adjusting its upper part and an ivory cone *J* attached to the bottom. The point of the ivory cone is the zero point of the scale *H*; for instance, 30 in. on the scale is exactly 30 in. from the point of the ivory cone. To the casing of the instrument there is attached a thermometer.

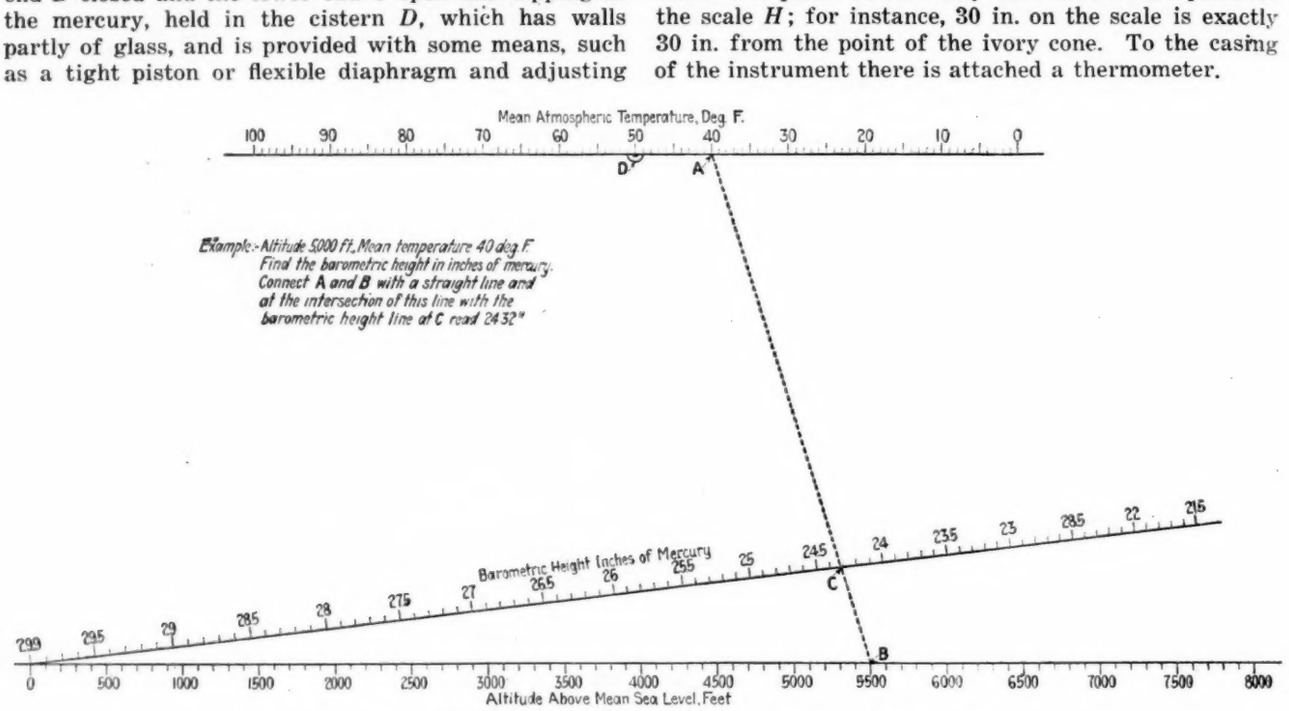

Mean Atmospheric Temperature, Deg. F.
100 90 80 70 60 50 40 30 20 10 0
D
A
Example:- Altitude 5000 ft, Mean temperature 40 deg F.
Find the barometric height in inches of mercury.
Connect A and B with a straight line and
at the intersection of this line with the
barometric height line at C read 24.32"
Barometric Height Inches of Mercury
29.9 29.5 29 28.5 28 27.5 27 26.5 26 25.5 25 24.5 24 23.5 23 22.5 22 21.5
C
B
0 500 1000 1500 2000 2500 3000 3500 4000 4500 5000 5500 6000 6500 7000 7500 8000
Altitude Above Mean Sea Level, Feet

FIG. 2—FINDING BAROMETRICAL PRESSURE ABOVE SEA LEVEL

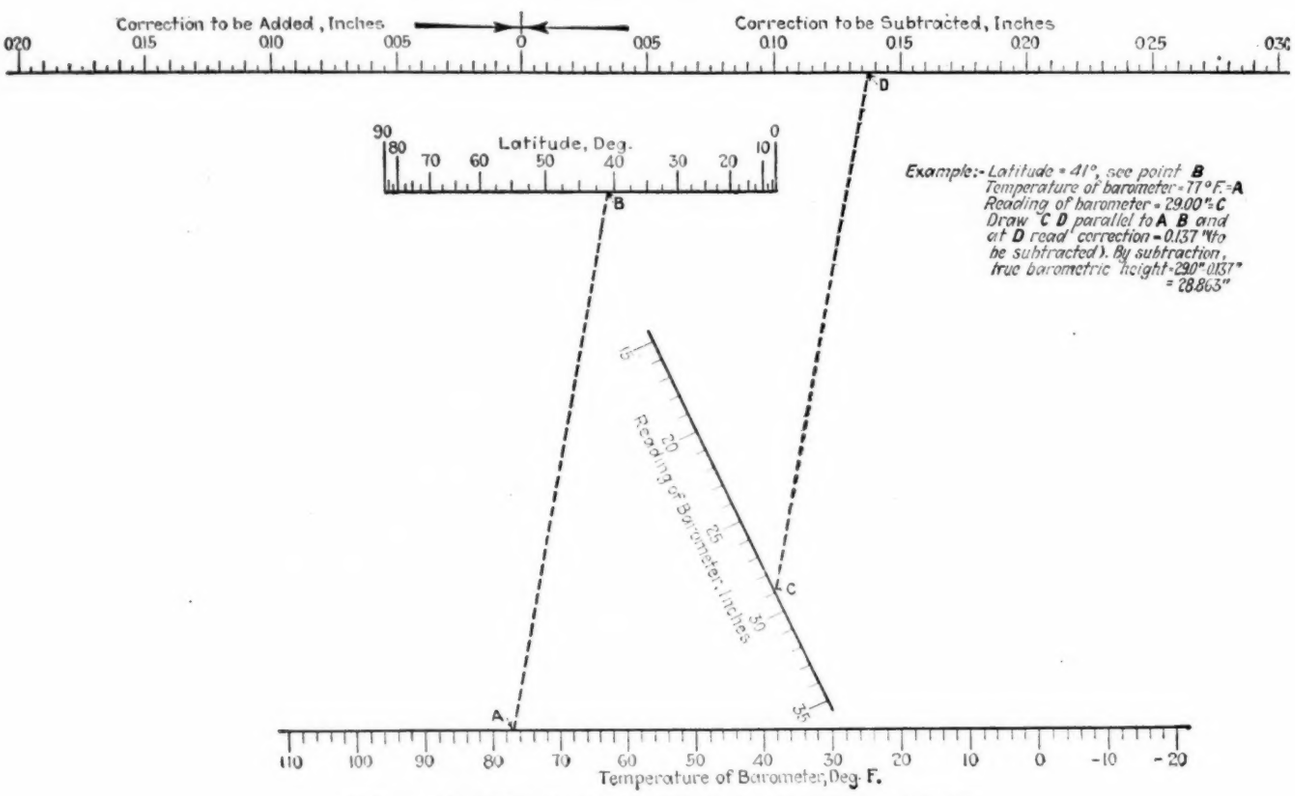

Correction to be Added, Inches
020 015 010 005 0
Correction to be Subtracted, Inches
005 010 015 020 025 030
D
90 80 70 60 50 40 30 20 10 0
Latitude, Deg.
B
Example:- Latitude = 41°, see point B
Temperature of barometer = 77° F. = A
Reading of barometer = 29.00" = C
Draw C D parallel to A B and
at D read correction = 0.137" (to
be subtracted). By subtraction,
true barometric height = 29.0" - 0.137"
= 28.863"
Reading of Barometer, Inches
15 20 25 30 35
C
A
110 100 90 80 70 60 50 40 30 20 10 0 -10 -20
Temperature of Barometer, Deg. F.

FIG. 3—CORRECTION FOR LATITUDE AND TEMPERATURE

screw *E*, for raising and lowering the level of the mercury. This cistern is inclosed so that mercury will not be lost in careful handling of the instrument. An air valve *F* is sometimes fitted, and is open when the instrument is being used. A brass casing *G* surrounds the glass tube, protecting it and carrying a scale *H* on

In setting up the instrument, it is first inverted and the tube completely filled with pure mercury. This is a delicate process and is done by the manufacturer or in a laboratory properly equipped. The constructon of the instrument is generally such that it can be inverted without the loss of mercury, by closing the air valve *F*.

Bringing the instrument carefully into an upright position, the mercury is seen to drop in the tube. There is a vacuum in the top of the tube, and the atmosphere, exerting its pressure on the mercury in the cistern, forces it up the tube until it is balanced by the hydrostatic pressure of the mercury. The purpose of the scale is to read this head.

To read the scale, raise or lower the level of mercury in the cistern D by the use of the adjusting screw E at the bottom until the ivory point J barely touches the liquid surface. By means of the thumbnut K the vernier L can be set with its lower edge even with the top of the curved mercury surface or meniscus, in the tube, and the exact reading of the scale can be observed on the scale H by using vernier L, which is similar to that on a vernier caliper.

After the observed barometric height and the temperature indicated by the attached thermometer have been noted, the next step is to ascertain what corrections must be applied.

The pressure of the atmosphere in pounds per square inch is equal to the weight of a column of mercury of

TABLE OF ALTITUDE CORRECTIONS FOR MERCURIAL BAROMETERS

Altitude Above Sea Level, Feet	Average Correction, To Be Subtracted From Observed Barometer Reading, Inches
0	0.000
500	0.001
1,000	0.002
2,000	0.003
3,000	0.005
4,000	0.006
5,000	0.008
6,000	0.009
7,000	0.010
8,000	0.011
9,000	0.012
10,000	0.012

the observed height and of one square inch in section. This depends on the density of the mercury in the barometer, which varies with temperature, and depends also on the strength of the force of gravity at the particular location, which in turn depends on the latitude and the altitude above sea level. The standard conditions for mercury are 32 deg. F. at a latitude of 45 deg. and a zero altitude—that is, at sea level. Under these conditions mercury weighs 0.4906 lb. per cu.in. The purpose of the corrections is to reduce the reading of the barometer to standard conditions so that the foregoing figure may be applied.

Without going further into the reasons for these corrections, the chart, Fig. 3, is presented for the purpose of quickly determining the temperature and latitude corrections of a mercurial barometer with brass scale. An example is worked out on the face of the chart, which is self-explanatory. This chart may be cut out, pasted on a card and hung on the wall by the side of the barometer.

In using the chart at a single station, the latitude will be constant. It can be ascertained by looking up the location on any good map, where the latitude can be read on the right or left margin of the map. The correct point on the latitude scale can then be marked for all future use.

There is another correction that is so small that it may usually be neglected, but is constant for any location. This is the correction for the decrease in the force of gravity at high altitudes. The table herewith gives, in the second column, these corrections for normal atmospheric pressures at the altitudes indicated in the first column.

For reference purposes, it might be noted that the chart and table are based on the following formula, used by the United States Weather Bureau:

$$B = B' \left(1 - \frac{t - 28.63}{10{,}978 + 1.112t}\right) (1 - 0.0026 \cos \phi)\ (1 - .0000000597\, h),$$

in which B is the corrected barometric reading; B' is the observed reading; t is the reading of the attached thermometer; ϕ is the latitude of the place; and h is the altitude of the place above sea level, in feet.

By multiplying the corrected barometric height in inches by 0.4906, the atmospheric pressure is reduced to pounds per square inch.

The Effect of High Overhead Charges

This is the third installment of a series of letters wherein two engineers discuss the Diesel engine. This one deals with the effect of high overhead charges when comparing the costs of power.

PITTSBURGH, PA., Sept. 23, 1923.

Mr. John Haite,
Worcester, Mass.

Dear Johnny:—When last I wrote, I promised to outline what I think real overhead charges should be. I have already given you my impression as to what is the useful life of a power plant, especially that of a Diesel-driven one; I am of the opinion that 25 years should be taken as the conservative life for the latter. Now the money invested in the machinery does not disappear instantaneously at the end of 25 years. To the contrary the engine is being worn out every day it runs. So that instead of suffering the capital loss in the year when the engine is junked, this wear should be added to the cost of making the factory product. One year the wear is greater than another, but we can say that on an average each year the engine's worth diminishes by $\frac{1}{25}$ of its first cost, or 4 per cent. Remember I wrote that I had found that in 10 to 15 years the engine would need an overhauling costing somewhere around 20 per cent of its first cost. If this overhauling was charged to that particular year, the power cost for that time would be excessive, so it seems to me that this should likewise be spread over the engine's life, which means an additional 1 per cent to the wear-and-tear charge to be added to the selling price of the product.

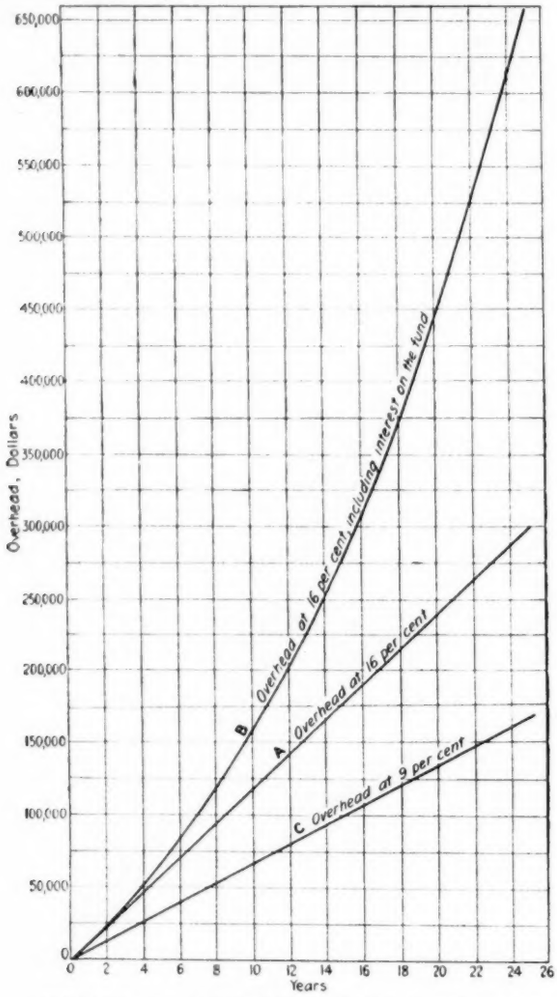
Taking a year's business as an example, we should, after figuring operating cost per horsepower or kilowatt-hour, which is added to the other factory costs, add 5 per cent of the initial power-plant cost to cover the wear-and-tear, or depreciation charge.

The money originally used to buy the power may be borrowed or taken out of the company's bank account. No matter where it comes from, the 5 per cent depreciation is banked or used to pay back part of the company's loan; in effect, at the end of the first year 4 per cent of the original investment is repaid to the capital account and an extra 1 per cent put aside for the overhauling or used to decrease the capital invested in the engine.

Money, whether borrowed or taken from a company's bank account, is worth the going rate of interest. Say this is 6 per cent. If 5 per cent is repaid during the first year, this portion does not bear any interest during

the second and succeeding years. Likewise 5 per cent is repaid the second year, etc. In effect, then, the net interest charge is decreasing each year and averages during the 25 years one-half of 6, or 3 per cent. The interest and depreciation is then 5 + 3, or 8 per cent.

Now as to insurance and taxes. Since the plant is wearing out each year so that in 25 years it is worthless, the amount of the insurance policy should be correspondingly reduced, as should also the taxing value. In other words, if the rate is 2 per cent, the average yearly rate based on the first cost becomes 1 per cent. The



EFFECT OF EXCESSIVE OVERHEAD CHARGES

total overhead then figures 9 per cent and not 16 per cent, as you originally assumed.

To show you the effect of using such an absurd overhead I have made up a chart. The assumption has been made that the plant capacity is 500 kw. and that it cost $150 per kilowatt or $75,000. As in most industrial plants, the load factor is taken to be 70 per cent and the plant is to operate twenty-four hours a day, six days a week. The hours per year are then 7,488 and the total yearly load 2,620,800 kilowatt-hours. Overhead at 16 per cent equals $12,000 per year, which figures out as 4.5 mills per kilowatt-hour.

Basing the usual life of 25 years on the grounds that over 98 per cent of the Diesel engines built 15 years ago are still in service and the fact that with an overhauling cost of 20 per cent of the first cost the engine can be made as good as a new one, the chart shows what the cumulative overhead charges at 16 per cent amount to during this period.

This curve marked *A* covers interest (6 per cent), depreciation (8 per cent), insurance and taxes (2 per cent). Bear in mind that since this sum of money, namely $12,000, is put aside each year, it is bearing or should be bearing interest. The curve marked *B* includes the compounded interest (6 per cent) on the overhead and shows that at the end of 25 years the power plant has been charged with a total of $658,380.

If, as I have pointed out, the real overhead is 9 per cent, curve *C* shows the amount properly charged against the plant, the total at the end of 25 years being $166,750. In other words, the use of excessive and unreasonable overhead charges would cause the power plant to be overcharged the sum of $491,630, or practically seven times the plant's initial cost. With a yearly output of 2,620,800 kw.-hr., or 65,520,000 in 25 years, the surplus depreciation amounts to 7½ mills per kilowatt-hour generated.

If you should decide to purchase power at the same price that your figures show it would cost if generated in the factory's plant, the use of the high overhead means you will be paying far too much for the purchased current. Even if you buy at ½ cent per kilowatt-hour less than the cost with your own plant, the high overhead means you are losing over $150,000, a not untidy sum.

My contention is that when comparing purchased and generated power one should use accuracy as to overhead. On the other hand, if you do put in a plant, it is advisable to set aside a liberal sum to cover depreciation, exactly as the premiums on one's life insurance are quite beyond the amount necessary to cover the actual losses due to death, etc. As with insurance, the premium not used to cover losses will be returned to you.

I hope my ideas may be of help to you.

With best wishes,

JIM.

Designing Power Plants by Impulse

In a recent letter, one of *Power's* good friends, a prominent designing engineer, made the statement that much of the present power-plant design had not even a good hunch behind it. Both the subject matter and the comments are of such general interest that the editor is impelled to publish the following extract:

"I was much interested in the editorial, 'Designing Power Plants by Impulse,' in the Oct. 2 issue. The only comment I would make is that I believe much of the present plant design is not even backed up by a good 'hunch,' but is the result of ignorance of the art, not even taking account of the known established factor of economy—both in investment and in operating results. I have in mind one plant now under way where for heat balance in one installation there have been installed house turbines, economizers and stage bleeding. How it is possible to use any more than two of these (and I consider even two inefficient) I do not understand.

"Referring to the last paragraph of the editorial, there are some people in the country who work along the lines indicated and when a free hand is given to a competent man it is bound to produce results. Nevertheless it must occur to all of us that it would be a dangerous experiment to permit some of those engaged in this line of work to have unrestrained run of things."

EDITORIALS

F.R. LOW, EDITOR

Heat-Treating Metal with Steam

STEAM is used for thousands of purposes, from pushing the pistons of engines to cooking candy, but our contemporary, the *American Machinist*, undoubtedly sprang a surprise on its readers when it told, in a recent issue, of heat treatment by steam. It appears that for some time a New England machine shop has treated certain small metal products in a bath of high-pressure saturated steam and water. Treatment takes place in the boiler itself, which is nothing but a short piece of four-inch extra-heavy pipe closed at one end. After introducing a pint of water, along with the parts to be treated, a plug is clamped on the open end and the steam pressure raised with gas jets. The pressure is brought up to about four hundred pounds and then released to atmospheric. Since the temperature of four hundred pound steam is only about 450 degrees F., it seems probable that the operation involved consisted of drawing the temper of hardened carbon-steel parts.

The idea is so novel that it is interesting to consider whether it is capable of wider application in this field. The temperature of 450 degrees obtained in the case mentioned would correspond roughly with the straw-yellow temper ordinarily given such tools as milling cutters. Softer tempers would require higher steam temperatures and pressures. For example, the brown-yellow temper of twist drills would require 500 degrees or about 700 pounds, the light purple of cold chisels 530 degrees or about 900 pounds, while the darker purple of woodsaws and screwdrivers would require 560 degrees or about 1,150 pounds. The colors themselves might be absent, but this is immaterial if the temperatures are right.

It then appears that below the extreme upper limit of present-day American boiler pressure (1,200 pounds) it is possible to draw the temper of carbon steel as low as is necessary for any ordinary tool or "tempered" machine part. There is apparently nothing to prevent such tempering being carried on in drums similar to those used on the new 1,200-pound boilers. With a little water in the bottom to insure a condition of saturation, and connected to a miniature high-pressure boiler, such a container would adjust itself almost instantly to the saturation temperature as the steam pressure was raised or lowered.

The questions at once arise: Why bother with such an unusual method? Are not present methods of tempering both convenient and satisfactory? Very probably so; the answer is not in the province of a power-plant journal. This is only an idea—perhaps an impractical one. At any rate the heat-treating experts should know that these steam pressures and temperatures, with inherently sensitive temperature control by pressure alone, are available for the asking if they can use them.

The same method might possibly be applied to the low-temperature annealing now widely used to relieve the internal strains of hard-drawn brass without serious loss of hardness. Brass rods and tubes are sometimes heated enough for this purpose by passing an electric current through them. The temperature may then vary a little as air currents affect the rate at which the heat supplied is removed from the metal. In the case of brass tubing the direct admission of high pressure saturated steam to the interior would apparently produce the same effect with a more definite temperature control. This, again, is an untested idea, that must be taken for what it is worth by those more familiar with the technique of brass manufacturing. One evident limitation would be the steam pressure the tubes could safely carry.

Welding Versus Flanges

HABITS of thought are peculiar things. For years engineers, when thinking of piping, have grown accustomed to the idea that two pairs of flanges the size of the main are necessary for each outlet, however small. Consistent adherence to this idea has often necessitated considerable expenditures for relatively small results. For example, a six-inch valved outlet in a sixteen-inch steam main requires (according to the old school of thinking) one sixteen- by sixteen- by six-inch cast-iron tee, one six-inch valve, two sixteen-inch gaskets and two six-inch gaskets. Yet the costly tee, with its sixteen-inch flange joints to be kept tight, could ordinarily be eliminated by the use of autogenous welding. It is no trick at all to cut a six-inch hole in the main and weld on a six-inch nozzle. In this, as in other examples of pipe-line welding, the most striking advantage of welding over the ordinary procedure is the partial elimination of flange joints, particularly in the larger sizes. This not only saves the first cost of flanges, gaskets and bolts, but simplifies the problem of keeping the lines steamtight.

Another important disadvantage of flanges is bound up with the problem of heat insulation. The insulation of straight runs of pipe with sectional covering presents no difficulty. When a flange is encountered, there are three possible courses. The flange may be left bare, but with a ten-inch steam line this means wasting about one ton of coal a year for each pair of flanges. Plastic insulation may be applied directly to the flange, but this will cost more than the same length of sectional covering, waste more heat and be subject to destruction each time it is necessary to get at the joint. The third course is to use a molded removable flange cover. This is perhaps the best procedure, although the cost is necessarily high and the heat loss greater than for straight insulated pipe. By getting rid of flanges wherever possible by the use of welding, these problems are largely avoided. Welding should be credited with any resulting reduction in heat losses and cost of insulation.

When, in addition, it is considered that all systems of autogenous welding, as well as gas cutting, are particularly adaptable to special conditions, it is not

surprising that welding is now widely used in the fabrication of power-plant piping. That it was not generally adopted years ago is due partly to human inertia and partly to the early imperfections of the art. For a number of years past every aspect of the welding question has been intensively studied by scientists co-operating with practical welders. So much progress has been made that any lingering prejudices are dying out. Whatever may be the proper viewpoint as to the welding of boilers, there seems to be little question as to the complete reliability of piping fabricated by competent welders.

The time has arrived when no engineer can afford to remain ignorant of the possibilities of welding in the power plant. In this connection the leading article in this issue, showing present-day standard practice in the fabrication of power-plant piping systems, is of timely interest.

The High Cost of Low Power Factor

LOW power factor and its bad effects form a subject that has been the center of a great deal of controversy both in this country and in Europe, yet the question is still with us and the solution is not yet in sight. Although the cost of poor power factor has not been properly evaluated to allow a general statement to be made on the economic phase of the subject, some figures on isolated cases show the losses to be high. The losses due to low power factor will vary with conditions, but those within the power plant are a small part of the whole. If the generators are designed for eighty per cent power factor, as they generally are, and the power factor of the system is maintained around this value, then about the only increased investment cost in the plant is for the twenty per cent in generator capacity. The prime movers, boilers and other equipment are designed for eighty per cent of generator capacity, therefore, in so far as these are concerned, they operate about as efficiently as if the power factor were unity.

It is outside of the generating station that the really large losses and increase in investment result from low power factor. On one power system where a careful study has been made, it was found that, owing to low power factor, the yearly power losses in generators, transformers, transmission and distribution lines were increased about seventy-one million kilowatt-hours over what they would have been had unity power factor been maintained. At a production cost of seven-tenths of a cent per kilowatt-hour this amounts to a direct loss of one-half million dollars a year. This power was produced and delivered to the system, and it would not have cost any more to have put it to revenue-producing purposes than to supply the increased losses in the system. In fact, it frequently costs less to utilize the power for the former, since the losses may involve an investment in regulating equipment to compensate for some of the bad effects of poor power factor. In a water-power plant water is passing through the wheels to produce energy that is utilized in the system to supply the losses due to bad operating conditions. In a steam plant expensive fuel is burned and converted into electrical energy that produces no revenue unless the customer is penalized for low power factor.

In this installation the investment in generating stations, transmission and distribution lines amounted to about five, twelve and twenty-seven million dollars respectively. Of this total investment over eleven million dollars was due to the power factor being less than unity, in this case about eighty per cent for the transmitted load, which is considered as fairly well up to standard. If fixed charges are taken at twelve per cent on the increased investment, the total increased costs due to energy losses and fixed charges are brought up to over one and three-quarter million dollars. If the increased energy loss in the customer's distribution systems had been included in these figures, the loss due to low power factor would be considerably higher.

Although these figures are for a large central-station system, they show that in industrial-power systems operating at low power factor the losses must be high with a resulting increased cost. Furthermore, the greatest losses do not exist in the power plant where the investment is the smallest, but in the transmission and distribution systems where the greatest investment is made. There is much to be said on the various phases of the power-factor questions, but the foregoing is sufficient to show that the whole problem is one of real economic importance and deserves the serious consideration of both the producers and the users of power.

Rounding up the Robbers

Since the coal strike was settled in September I have taken up the question of price in a series of conferences with the members of a committee of the operators. On Friday last this committee finally declined to recommend to the coal industry that it should clean its own house of extortion, as I have been urging it to do.

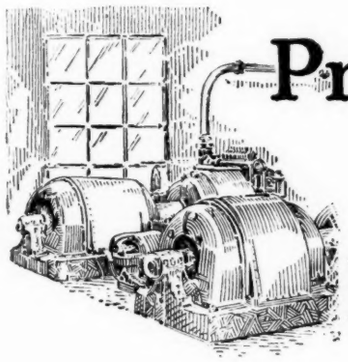
We do not have to prove the existence of abuses in the anthracite industry or of crying injustice to the consumer in extortionate prices. They are matters of common knowledge. It would have been merely the most elementary common sense for the committee of operators to take these evils in hand. Being the producers of the coal, they have the power to do so most effectively. They could, if they would, not only give the consumer a square deal, but at the same time free their industry of the public condemnation which now rests upon it. They have declined to do so. Then we must turn elsewhere.—*Governor Pinchot.*

More power to you, Governor! With laws that made extortion a prison offense or put it on a par with less polite methods of stealing and officials like this to enforce them, the price of coal would come down to where it belongs.

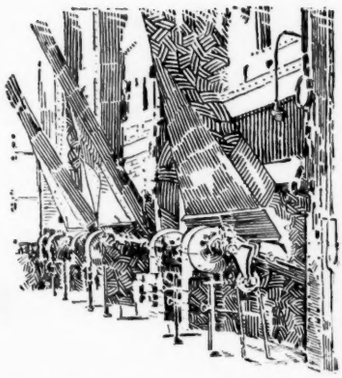
With a total of over thirty-five hundred boilers in use Queensland has been entirely free from fatalities in connection with their operation during the last year. This enviable record can be attributed largely to the rigid enforcement of the "Inspection of Machinery Act" in that province. Some of our states might well take note.

A well-known engineer, observing that many of the stoker manufacturers have hedged their bets by annexing powdered-fuel systems to their lines, suggests that it might not be a bad plan to pick up a system or two for coal gasification while they are still selling cheap.

Another illustration of Congressional logic on finance! Representative Madden proposes that the government build a new plant to replace that at Gorgas, recently sold to the Alabama Power Company, in order to meet Mr. Ford's offer of five million dollars for the whole project.

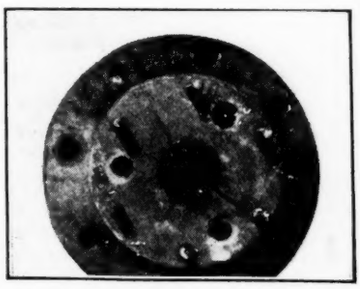


Practical Ideas *from* Practical Men



Scale Causes Cylinder-Head Fracture

Engineers in charge of oil-engine plants do not give enough attention to scale in the cylinder jackets and heads. If the cooling water contains much scale forming material the scale will settle out as the water warms up. Since the head is the hottest portion of the engine, more scale forms here than in any other part of the jacket. It is by no means unusual to find plants where new heads are needed every few weeks. The writer was called in by the owners of a plant of this type and found over a dozen cracked heads scattered around the place, one being shown in the illustration. Investigation revealed that the water was very bad, and as a remedy I suggested that rain water from an unused cistern be used. This, being free of all mineral, would cause no scale. To procure a supply of makeup rain water, the roof of the plant was equipped with guttering and the rain so caught was led to the cistern. Since then no heads have given trouble.



CYLINDER HEAD FRACTURED DUE TO SCALE

St. Louis, Mo. J. SPUGG.

Testing the Boiler in the Small Plant

A boiler test for evaporation is such a simple procedure that the practice of periodical testing of boilers for evaporation and efficiency should be more generally adopted by men in charge of isolated plants. The evaporation test is valuable in that from its results the boiler efficiency and the cost of steam can be computed. At best a boiler is an inefficient piece of equipment, and any effort expended to raise this efficiency is time and energy well spent.

Knowing definitely what the boilers are doing should be a part of every engineer's duties, and in order to figure the cost of isolated plant power, this factor must generally be known. In plants equipped with water meters, recording thermometers and coal-weighing equipment the process is somewhat simplified, but the small plant without these refinements can obtain satisfactory results.

To run an evaporation and efficiency test, the following items must be known: Total weight of coal burned during the test, total weight of water evaporated during test, average temperature of feed water, average steam pressure, and the heat value in B.t.u. of the coal used. The test should be run for a period of not less than eight hours. A longer test will give more accurate results.

In the absence of coal-weighing equipment the best method of weighing the coal is to use a platform scale and weigh each wheelbarrow load. To weigh the feed water, the simple method of using two barrels, one elevated above the other, may be used. The feed water is piped to the upper barrel and is discharged into the lower one through a valve placed in the bottom of the barrel. The lower barrel connects to the suction of the feed pump. It is advisable to have a bypass to the feed pump from the regular water supply in case something goes wrong with the barrel arrangement. Weigh the upper barrel when empty, then fill with water and weigh again. In this way the weight of water used during the test may be determined by multiplying the weight of water in the barrel by the number of times the barrel is filled.

In order to start the test, make a note of the condition of the fire, so that it may be left in the same condition when the test ends. If the boiler was fired five minutes before the test started, see that it is fired five minutes before the test ends. It is only in this way that the same conditions will exist at the start and finish, and the coal weighed will be the same as the coal consumed.

Note the level of the water in the gage glass and mark this level with chalk or a cord. When the test is nearing completion, see that the water is at this mark. If there is no thermometer in the feed-water line, suspend one in the upper barrel of water. The feed-water temperature and the pressure on the boiler should be read every half hour. If either of these fluctuates considerably, it is better to take readings every fifteen minutes. At the end of the test add the readings and divide the sums by the number of readings; in that way the average temperature of feed water and steam pressure will be obtained.

In order to show how the evaporation test is figured from the results obtained, the following case is worked out:

The weight of coal burned, 4,400 lb.; total weight of water evaporated, 32,000 lb.; average temperature of feed water, 184 deg. F.; average steam pressure, 110 lb.; length of test, 8 hours. From the results obtained it is found that 4,400 lb. of coal were used and 32,000 lb. of water evaporated. Hence $32{,}000 \div 4{,}400 = 7.2$ lb. of water evaporated per pound of coal burned. This 7.2 lb. is the "actual evaporation." The "equivalent evaporation from and at 212 deg. F." must now be calculated.

It is a well known fact that it takes more coal, and therefore more heat, to make steam with cold water than with hot water, also that it is easier to make steam at a low pressure than at a high pressure. Therefore it is obvious that the heat required to evaporate 7.2 lb. of water into steam will depend on two things: the temperature of the feed water, and the pressure of the steam in the boiler.

In order to change a pound of water at 212 deg. into steam at 212 deg. F. at zero gage pressure, there must be added to it 970.4 B.t.u., which is called the latent heat of steam at atmospheric pressure. In the example given, the feed water temperature was 184 deg. F. and the steam pressure 110 lb. gage. The amount of heat required to change a pound of water at 184 deg. into steam at 110 lb gage is not the same as that required to make a pound of steam from and at 212 deg. Therefore the actual evaporation must be multiplied by a "factor of evaporation."

The factor of evaporation for the temperature and pressure given is found from any table of factors of evaporation to be 1.07. The factor 1.07 indicates that to change a pound of water at 184 deg. into steam at 110 lb. pressure requires 1.07 times as much heat as to change a pound of water at 212 deg. F. into steam at atmospheric pressure. Hence, $7.2 \times 1.07 = 7.7$ lb., equivalent evaporation from and at 212 deg. F.

As it takes 970.4 B.t.u. to make a pound of steam from and at 212 deg., to make 7.7 lb. will require $7.7 \times 970.4 = 7{,}472$ B.t.u.

The efficiency of the boiler may be expressed as a ratio of the output to the imput. By output is meant the quantity of water evaporated per pound of coal fired. Input is the amount of heat contained in a pound of the coal as fired. It is preferable to have a test made to determine the heating value of the coal used during the test. If this cannot be done, then learn the grade of coal and the district from which it came. Many handbooks on engineering contain a table giving the heating value of various grades of coal used for steam power purposes.

In the case worked out in the foregoing it is assumed that the coal has a heating value of 13,000 B.t.u. per pound. For every pound of coal fired, 13,000 heat units are liberated, and this value is taken as the imput. The efficiency of the boiler may then be calculated as follows:

$$\text{Efficiency} = \frac{\text{output}}{\text{input}} \text{ or } \frac{7{,}472}{13{,}000} = 0.574, \text{ or } 57.4 \text{ per cent.}$$

Knowing what the boiler will do from the results obtained, it is a simple matter to figure the rating at which the boiler was operated during the test. The normal rating of a boiler is given as 3.45 lb. of water per square foot of heating surface per hour from and at 212 deg. F. Therefore the boiler rating developed, per cent, equals equivalent evaporation per square foot divided by 3.45 and multiplied by 100. Or, taking the figures given in the test, the factor of evaporation was found to be 1.07. Now, dividing 3.45 by 1.07 we get 3.22, which is the rated evaporation per square foot per hour instead of 3.45 as under the standard condition. Now say the boiler had 1,000 sq.ft. of heating surface, the evaporation at normal rating would be $1{,}000 \times 3.22 = 3{,}220$ lb.

Since the boiler evaporated $\frac{32{,}000}{8} = 4{,}000$ lb. of water per hour the rating would be $4{,}000 \div 3{,}220 = 1.24 \times 100 = 124$ per cent, or 24 per cent above normal rating.

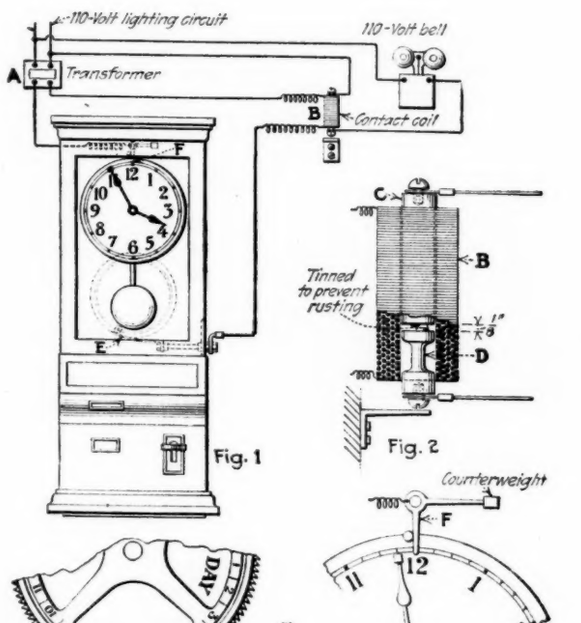
Louisville, Ky. A. R. KNAPP.

Improvised Automatic Time Signal

In order to provide an alarm or signal for several different services around our plant, we worked up the system shown in the illustration.

An old 50-watt potential transformer *A* was connected to the lighting service in the plant, as also was a 110-volt bell with the iron core of the contact coil *B* inserted in the circuit. The secondary voltage of the transformer is about ten volts, and this is connected to the winding of the contact coil through contacts in the time clock, as shown in Fig. 1.

A detail of the contact coil *B* is shown in Fig. 2. The coil is made by winding about 300 turns of No. 18 magnet wire on the fiber sleeve from a 30-ampere 600-volt cartridge fuse. A soft iron core *C* is made to fit



FIGS. 1 TO 4—ASSEMBLY AND DETAIL OF SOLENOID SWITCH AND CONTACTS

snugly into the upper part of the coil with a loose fitting spool *D* of the same material at the bottom and provided with a stop to prevent it from dropping too far away from the core *C*. As the parts *C* and *D* constitute the switch on the bell circuit, the inner ends of the cores are tinned to provide a good contact and prevent rusting.

As our clock is equipped with a brass wheel about ten inches in diameter with a slot turned into it near its outer edge for fastening metal lugs at any point desired, a brass strip *E* was attached to the side of the clock case as shown in Fig. 3. The upper contact is made by swinging the piece *F*, Fig. 4, so that it will make contact with the minute hand for ten to fifteen seconds. The length of the contact on the bell crank *F* determines the length of time the bell will ring. The minute hand of the clock is tipped with copper to prevent burning.

With this arrangement, when the circuit to the coil *B* is closed by the contacts *E* and *F* on the clock the

core *D* is drawn up against the stationary core *C*, thus completing the bell circuit.

The mechanism in the lower portion of the clock can be adjusted to ring the bell at any hour desired, the usual arrangement being for the starting and quitting time for workmen, regular shift for operators, time for taking log readings, etc. In our plant we use an eight-inch gong that can be heard practically all over the building, covering about two acres.

The employment of a gong obviates the necessity of a man's hanging around for ten minutes or more, often putting off important work for fear he will neglect to ring in on time, and even then it is often overlooked and he starts work several minutes late. With the employment of the automatic signal the usual irregularity is removed.

HERBERT HUGHES.

Burley, Idaho.

The Newcomen Engine

A short time ago, when in Edinburgh, I visited the Royal Scottish Museum and was much interested in a recently added exhibit, a model of a Newcomen engine, and from inquiries made and with the help of a local paper, I was able to obtain an interesting history.

This engine, from which the model had been made in the museum workshops, was originally erected by a John Macintyre in 1810 at Farme Colliery, Scotland, and represented the highest development of the Newcomen engine. After working continuously for 105 years, it was taken down and presented to the Glasgow Corporation, and being the last to work in Scotland and probably anywhere, Glasgow has obtained a valuable relic for preservation.

It has often been said, and possibly quite truly, that a much larger share of the invention of the steam engine has been awarded to James Watt than was due, much to the disadvantage of Newcomen and his partner, Savery.

Newcomen was born seventy years before Watt, and very early in life turned his genius to the problem of pumping water from the mines. At that time the only power available was horse power or, as often the case, man power, but as the quantity of water in some districts increased at an alarming rate, many mines had to be closed up altogether. This problem had been taken up by Thomas Savery a few years earlier, and he patented a successful but highly dangerous "fire engine." Newcomen must have been engaged upon his invention about the same time, as about 1712 he brought out his engine, from which it may be truly said all subsequent steam engines directly descended.

The original features that it is claimed he introduced were a separate boiler, a cylinder with a piston working therein and the provision of valve gear to allow the engine to work automatically. His engine was also safe, as he filled the vertical cylinder with steam at practically atmospheric pressure, then sprayed cold water into the cylinder to condense the steam and form a partial vacuum. The pressure of the atmosphere on the top of the piston then forced it down, causing the engine to work.

Savery's patent being sufficiently wide to cover all engines using steam, he wisely ceased building "fire engines" and joined Newcomen in manufacturing the atmospheric engine, which was so successful that in a few years over a hundred and fifty were working in mines in the north of England and Cornwall. And so skillfully had Newcomen designed his engine that it existed for over seventy years in its original form, until it was gradually superseded by the engines of Watt, which were certainly much more economical.

In the making of the original engine referred to, a very interesting document is in existence, which shows the materials used in its construction. It does not state where the castings were made, but they were taken by road to London and shipped along with the elms for the pumps to Newcastle, where they gathered up the lead for making the boilers and more timber, the whole being then carried on to Leith, Scotland. Two workmen at fifteen shillings each per week took twenty weeks to erect the engine, while the contractor from England received fifty pounds for his "pains of coming and going upon account of Edmiston Engine," evidently a similar engine that was already working in Scotland which he was to copy, and the two workmen already referred to also received three pounds each for "coming and going," which suggests that they also came from England.

There are many other interesting items of payment, and the total cost of the engine "erected and working" was slightly over one thousand pounds, with the addition of eighty pounds per annum payable to a committee in London for a "license to work." This committee appears to have been appointed and authorized by the proprietors of the "invention for raising water by fire," but it is not recorded whether or not they continued to draw their royalties right through the period of 105 years that this engine was working, and more information on this point would, I am sure, be of considerable interest.

F. P. TERRY.

Belfast, Ireland.

Flue-Gas Thermometer as Substitute for CO_2 Instrument

It is probably the wish of most engineers to have a recording CO_2 instrument, with which to check boiler performance, but sometimes the cost makes it hard to get the approval of the powers that be.

We have been trying out recording flue-gas thermometers and find that results in small plants compare quite favorably with CO_2 instruments. We know that with a clean boiler, good baffles and settings and with 13 or 14 per cent CO_2 the excess air is cut to a minimum and the temperature of the flue gas will be low; that is, from 100 to 200 deg. F. above the temperature of the water in the boiler, depending on the load. The temperature of the water in the boiler can be found from the steam tables. At normal rating there should not be over 150 deg. difference. If we admit an excess of air, the flue-gas temperature will immediately rise and the more the excess the higher it will go.

With a long-distance recording thermometer installed at about one-third the cost of a CO_2 instrument, we have a fairly good check on the condition of the boilers and how the firemen handle the fires. If the temperature goes up, there is probably one of three things the matter—an excess of air, dirty tubes or leaky baffles. The CO_2 instrument might show good CO_2 and still the tubes could be dirty or the boilers have leaky baffles. Of course, for best results the instrument should be placed where the firemen can see it at all times. We use the recording thermometer on both oil- and coal-fired boilers and get equally good results.

Palatka, Fla. C. H. COURSER.

Comments from Readers

Breakage of Connecting-Rod Wedge Bolt

Referring to the inquiry of S. P. in the Oct. 30 issue as to the breakage of connecting-rod wedge bolt, may I offer a suggestion as to the probable cause of this trouble? In the operation of one of our engines we had considerable difficulty in keeping a bolt in place, and it was not uncommon for us to have three or four extra bolts on hand. It was impossible to look for the trouble at the time, the only thought being to keep the engine in service until cleanup.

We took the brasses and wedge out of the rod and everything seemed to be O.K. Upon trying the bolt in the wedge, we noticed that the unthreaded part fitted loosely in the wedge, which was threaded only part way through. This play was hardly noticeable, nevertheless we concluded it to be sufficient to cause the trouble, so we ordered a new wedge immediately, and when it arrived the same bolt was tried in it and was found to be a proper fit. We put this new wedge in and haven't broken a bolt since. PIERRE T. WILLIT.

Paulding, Ohio.

Catalogs and Advertising for the Boys

I failed to read Mr. Terry's letter entitled "Catalogs and Advertising for the Boys," when it appeared in the Aug. 14 issue, but read it after having read E. W. Brong's letter in the Oct. 9 issue with reference to the same subject. These two letters contain what I consider some of the best information and advice that could be written. I heartily indorse both of them.

I am a young engineer, perhaps not so far advanced in the profession as many readers of *Power*, but it gives me pleasure to state that the things that have meant the most to me are those I learned from my first "boss" and from reading literature given me by men higher up who were kind enough to pass such information along. I began work at the age of thirteen years. I chopped grass, cut weeds and rubbish around the plant, swept out the engine room and cleaned the machinery. The superintendent and the day engineer were men who enjoyed seeing a young fellow work up, and as I worked, they showed and taught me many lessons and facts that were as sound an engineering foundation as a beginner could want. I shall always remember these two men most kindly and shall forever be indebted to them for experiences and lessons that have meant much to me.

Not so many years ago I was one of the operating engineers in a small plant. A traveling salesman representing one of the largest concerns in existence today called and asked for the superintendent. Inasmuch as the superintendent was out, I endeavored to show the salesman every courtesy. I shall never forget the contempt with which he regarded me. It was certainly anything but becoming to a high-class salesman.

A few months ago I had occasion to give quite a nice order, and it just so happened that this same salesman called to investigate. He did not remember me, but I shall never forget him. His competitor made as good a proposition on the order as he did, and needless to say he got it.

Not only in the engineering world, but in every walk of life, a kind word, thought or deed goes a long way.

E. S. BEE, Chief Engineer,
Sanatorium, Miss. Mississippi State Sanatorium.

Safety of the Steam Accumulator

I have read with interest the article "The Steam Accumulator and Its Applications" in the Aug. 28 issue, which gives a clear and concise account of the "Ruths" accumulator. One small correction I should like to make regarding Fig. 2, page 323, namely, that the arrows showing the direction of circulation of the water should be reversed as the steam nozzles point obliquely upward in these tuyeres.

In an editorial on page 335 of the same issue *Power* makes further mention of the accumulator and its possibilities and refers in particular to the explosion hazard.

Now without going into the question of whether it is quantity or quality which tells most in explosions (the accumulator has the former and a high-pressure water-tube boiler has the latter, like gunpowder and TNT for example, the boiler being the TNT), I should like to point out that, like steam boilers, these accumulators can be calculated and built to withstand the desired pressure. This usually ranges from an initial fully-charged pressure of between 70 and 140 lb. to a discharged pressure of 8 to 55 pounds.

They are recognized by the steam-boiler inspection authorities in most countries and subjected to the same tests as are steam boilers for the same pressures. It is nothing new to industry to have energy in such concentrated form.

Many boiler houses have rows of boilers containing a total volume of water in excess of that represented by the accumulator and under much more onerous conditions—that is, higher pressure, rapid temperature variation, etc.—and if one boiler explodes the whole row is likely to follow suit. Again, in pulp mills high-pressure digesters containing 350 tons of *acid* are by no means uncommon. Here again is an enormous potential energy complicated by the presence of acid.

An accumulator therefore calls for no specially strong nerves on the part of its owner. The conditions under which it works are much simpler than those of a boiler or digester, and its shape is ideal for withstanding pressure; that is, cylindrical with hemispherical ends. Compare such an apparatus with any type of boiler with its stays, pressed tubes, etc., and sudden cooling when cold air is suddenly admitted into the furnace.

The only danger lies really outside the accumulator and is that of priming or water hammer if the steam main from the accumulator should be ruptured, say by

an accident in the factory, and this danger is foreseen and guarded against as explained below:

Over one hundred installations are in operation, and in no single instance has water hammer occurred in the accumulator. This danger has been obviated by the nozzle-shaped outlet branch, which is so constructed that the maximum flow of steam through it is practically constant irrespective of the extent of the difference of pressure between the outlet pipe and the accumulator. In other words, supposing an accident should occur in the factory, causing the outlet pipe to break right off so that the full accumulator pressure acted upon a pipe *open full bore to the atmosphere*, the steam flow from the accumulator would not exceed this predetermined maximum for which the nozzle is constructed, the accumulator would discharge dry steam as usual and no water hammer could occur.

Special experiments have been carried out to discover the maximum rate of discharge permissible from an accumulator. At Holmens Bruk, a large paper mill in Sweden, an accumulator with a volume of 3,000 cu.ft. capable of discharging a total of 17,000 lb. steam with a drop of pressure from 85 to 28 lb. per square inch was specially tested for rapid discharge. A rate of discharge equal to 220,000 lb. per hour was maintained for a few minutes. The steam remained dry as usual, and no trembling occurred in the accumulator.

Stockholm, Sweden. NILS FREMNE,
Aktiebolaget Vaporackumulator.

Advantages of Welded Pipe Joints for Higher Temperatures

In the issue of June 5, 1923, there was a rather important editorial entitled, "How Higher Temperatures Affect Pipe Fittings." The statements made are conservative, and one in particular seems to cover the whole subject. It is as follows: "Evidently, one result will be to avoid, as far as possible, bolted joints for temperature extremes." It may be that the intention was to refer to fusion welded joints instead of bolted ones, although it is not clearly so stated, but if this solution of the problem was in mind, I suggest that it be given careful consideration because of its many advantages.

Commercial pipe, even of standard weight, has enormous strength, especially in the smaller sizes, because of the necessity of providing sufficient material at the bottom of the thread. This amount is not very great in standard-weight pipe, in the case of 4 in. the thickness at the bottom of the thread at the end of the perfect thread being about 0.13 in. The thickness of the pipe itself is 0.237 in.

The bursting strength figured by Barlow's formula, as given in *Machinery's* Handbook, is over 5,200 lb. per sq.in.; for extra-heavy pipe, nearly 7,500 lb. per sq.in., while the thickness of the latter is 0.337 in. If this size is about the maximum that is feasible in connection with the high boiler pressures now proposed, which limit is suggested in your editorial, there is no reason why installations of such pipe should not be made entirely by fusion welding.

The thickness of extra-heavy pipe is about $\frac{3}{8}$ in. and is one which is easily and safely welded. Welding materials have been developed that will easily give more than the strength of the pipe. The fiber stress on the circumferential joint is only half that on a longitudinal joint. There is no difficulty in getting or training welders for this work, and the testing of the welds is readily done; also the removal of welding strains is easily accomplished and any lack of alignment of pipe, due to the weld shrinkage, is easily removed. So it would seem that as far as the pipe lines themselves are concerned, there is no difficulty in providing a safe, flexible and strong pipe line in any case when needed.

The question of fittings and valves for such pressures and temperatures as are referred to in the editorial, must also be considered. Even in welded pipe lines at the present time fittings are of the usual bolted type and the pipes connected to them are also provided with bolted flanges of some approved design. In the cases under consideration the material is of necessity cast steel. There is no reason why extensions cannot be cast on such fittings, of a thickness that will be suitable for welding and of a length that will remove the weld far enough from the body of the fitting. This is particularly necessary with valves to avoid any distortion due to heating from the welding. Such fittings have not been made in the past, because there was not sufficient demand for them. If lines of small pipe have to be run in parallel in order to obtain sufficient capacity, the headers for such lines can also be made with extensions for welding.

There seems to me to be no reason why such installations cannot be made entirely welded. The reduction of weight, increased flexibility, the absolute tightness, the freedom from flanged joint troubles, are most advantageous. If it be objected that welding makes it difficult to replace or repair a pipe line, it is believed fair to reply that welded pipe lines need no repairs, and in case of necessity there is no more trouble or expense connected with welding in a tee or valve of proper design than in doing the same work with flanged joints. I further believe that until welding is used for the construction of such high-temperature pipe lines, there will be much difficulty, and I urge the early consideration of the proper design for this work, which is undoubtedly close on us. S. W. MILLER.

Long Island City, N. Y.

Expansion in the Steam-Engine Cylinder

In the article, "Expansion in the Steam-Engine Cylinder," in the Oct. 9 issue, Mr. Barker suggests that "re-evaporation" takes place in a steam-engine cylinder more prominently during exhaust than during expansion.

In an article on page 336 of the Aug. 29, 1922, issue, I suggested that particular idea as possibly underlying the economy of a unaflow engine, the idea being that the shortening of the exhaust period—a feature of the unaflow engine—decreased the opportunity for the transfer of the heat from the cylinder walls in time for rejection. This would not be the case if the transfer occurred during expansion.

Mr. Barker's article is good not only in the special matter it treats of, but it is useful in once more pointing out how varying conditions cannot be compared if related conditions are not the same. Steam leakage, unsuspected, has not only been a very frequent source of loss, but a misleader as to other conditions.

I should like also to commend the article on "Vacuum Heating" by J. E. Seiter in the Sept. 11 issue, because it really explains what he is trying to impart. Not knowing much on the subject myself, I obtained practical bottom ideas from the article. J. C. MORRIS.

New Orleans, La.

QUESTIONS AND ANSWERS

Conducted by Franklin VanWinkle

Adapting Grate Area to Boiler Load

We have a water-tube boiler with 3,500 sq.ft. of heating surface, but the maximum output is only about two-thirds capacity. The furnace is hand fired with shaking grates, and there is about 10 ft. clearance between the grates and tubes. The flue-gas analysis averages 10.6 per cent, but at each firing we have considerable smoke. With alternate firing there is 2 to 3 per cent less CO_2. The grates are in eight sections. Would it not be advantageous to brick over two of the sections to cut down the grate area to correspond with the load. J. F. J.

Assuming that the present grate area is proper for full capacity, reduction of grate area as proposed should result in greater activity of combustion with higher temperature of the fuel bed, more perfect combustion and less smoke, provided the stoking is uniform, with about three-fourths the present intervals between times of firing.

Water Required for Condenser Assisted by Closed Heater

How many gallons of condensing water will be required per minute by a jet condenser for a compound condensing engine developing 1,200 hp. with 26-in. vacuum, assuming steam consumption of 14 lb. per hp.-hr., when there is a closed heater in the exhaust pipe, between the engine and condenser, that raises the temperature of 20,000 lb. of water per hour from 70 deg. F. to 100 deg. F.? The temperature of the injection water is 70 deg. F., and temperature of the hotwell is assumed to be 110 deg. F. J. E. S.

Development of 1,200 hp. with consumption of 14 lb. of steam per horsepower-hour would be $1,200 \times 14 = 16,800$ lb. of steam per hour, or 280 lb. per minute, and to heat 20,000 lb. of water per hour, or $20,000 \div 60 = 333.33$ lb. per min., from 70 to 100 deg. F. would absorb $333.33 \times (100 - 70) = 10,000$ B.t.u. per min., and each pound of steam would be deprived of $10,000 \div 280 = 35.71$ B.t.u.

If the steam in the low-pressure cylinder at exhaust is dry and saturated, the heat of the steam as discharged will correspond to the total heat in steam at the exhaust pressure, but it usually contains considerable moisture, part of which is re-evaporated when the exhaust valve opens. Strictly speaking, the quality of the steam should be taken into account. However, it is sufficiently accurate for most practical purposes to assume the exhaust entering the condenser, or an intermediate heater, to be dry and saturated and its heat to correspond to the pressure.

The total heat of one pound of steam at 26 in. vacuum is 1,114.5 B.t.u. above 32 deg. F., and as the heater would remove 35.71 B.t.u., the heat per pound of steam presented to the condenser would be $1,114.5 - 35.71 = 1,078$ B.t.u. above 32 deg. F., so that each pound condensed to the temperature of the hotwell would have to part with $(1,078.79 + 32) - 110 = 1,000.79$ B.t.u.

As each pound of injection water would absorb $110 - 70 = 40$ B.t.u., each pound of the steam as presented to the injector would require $\frac{1,000.79}{40} =$ 25.02 lb. of condensing water, and for 280 lb. of steam per minute the amount of condensing water required would be $25.02 \times 280 = 7,005.6$ lb., or $7,005.6 \div 8\frac{1}{3} = 840.7$ gal. per minute.

Water in Air Bottles for Starting Diesel Engines

In our Diesel-engine plant there always is considerable water in the compressed-air bottles. Is this on account of leaks in the intercooler coils, and may it not interfere with good operation of the engine? E. D.

The water is due to the vapor pressure of moisture in the air. When a cubic foot of free air is at the temperature of 60 deg. F., the maximum pressure that can be exerted by the vapor is 0.25 lb. per sq.in. absolute, and the maximum weight of water vapor that can be present is 0.0008 lb. As the temperature is increased, the vapor pressure is raised and the weight of vapor that may be present is changed. The weight of vapor that may be present per cubic foot of air is as follows: At 60 deg. F., 0.0008 lb. per cu.ft.; at 80 deg. F., 0.0015 lb.; 100 deg. F., 0.0028 lb.; 150 deg. F., 0.0103 lb.; and at 200 deg. F., 0.0298 lb.

By action of the compressor the air taken in is reduced in volume; for example, 100 cu.ft. of free air at 60 deg. F. compressed to 900 lb. per sq.in. absolute, pressure would have a volume of $100 \times 14.7 \div 900 = 16.3$ cu.ft., assuming that the intercoolers and aftercoolers bring the temperature back to the original 60 deg. F. If the initial air contained 0.0008 lb. of water vapor per cubic foot, 100 cu.ft. of suction air would contain 0.08 lb., but as the new volume would hold only $16.3 \times 0.0008 = 0.01304$ lb., the difference of weight, or $0.08 - 0.1304 = 0.06696$ lb., will settle out in the discharge pipe and receivers. However, the amount of water vapor never is the maximum; that is, the humidity is under 100 per cent, so that the weight of water that would be thrown down per 100 cu.ft. of free air would be less. Failure to remove the water by bleeder valves may interfere with starting the engine. At times when moisture reaches the fuel valves, the engine will pound due to the water delaying ignition of the fuel after it enters the cylinder, for then the ignition occurs after the crank is past dead center.

Electrical Devices for Paper Machines

Why do modern paper mills use a separate engine to each machine for driving a direct-current generator that, in turn, supplies current to a variable-speed direct-current motor connected to the mainshaft of the machine, in place of obtaining power for several machines from one large engine through an electrical drive?

S. B. R.

Separate engines may be used for the purpose of obtaining closer speed regulation. However, the present tendency is toward complete electrical drives with the power generated in a central electrical plant, which for each machine supplies power to a constant-speed motor for the constant-speed end of the machine, and to an alternating-current motor direct-current generating set that in turn supplies current to an adjustable-speed direct-current motor connected with the variable speed line.

The special feature of the direct-current generator is that it is separately excited by a small exciter. By varying the current supplied to the main generator field, the delivered voltage can be varied, and in this manner the speed of the motor driving the machine can be varied over a wide range and the motor speed above normal can be obtained by means of field control. Combination of these methods provides a speed range of about 10 to 1.

To meet the variations in relative speed that may be desirable in different parts of a paper machine, a new development is to drive the rolls with several motors instead of one. By this system each section of the variable-speed line of the paper machine is driven by an individual direct-current adjustable-speed motor, connected to the shaft of each section by a reduction gear. This system not only simplifies transmission of power to the rolls but permits extremely accurate control. An automatic speed regulator is employed which controls the speed not only of the whole variable-speed line but of each motor also, so that different relative speeds can be obtained by means of which speed variations as small as one-tenth of one per cent are instantly detected and corrected.

Auxiliary Outside Belt

We have a troublesome belt drive consisting of a 90-in. diameter pulley driving a 20-in. diameter pulley through a 24-in. double leather belt with pulleys approximately 21 ft. centers. To transmit the maximum load of about 135 hp., it is necessary to keep the belt very tight, which we believe to be the cause of the laps of the two plies of the belt continually coming apart. One of our operators claims that the difficulty could be overcome by running a second belt ouside of the one now in use, his theory being that the outside belt could be run as a reasonably tight belt and the inside belt could be run quite slack; and that the outside belt would cause the inner belt to grip the pulley much better, thus causing the inner belt to pull most of the load. How would the proposed arrangement work out, and would deterioration of the belting be appreciably greater than in ordinary belt drives?

J. H. B.

Separation of the plies of the belt results from bending the belt around the 20-in. diameter pulley, combined with tendency of the outer ply to move at higher velocity and tear itself away from the inner ply, especially as the belt leaves the 20-in. receiving pulley. An independent single-thick belt placed around the outside of the present belt would move at higher velocity than the latter, but two such belts would work properly if kept apart except where they pass around the pulleys.

Regulation of the belt tensions cannot be solely depended upon for this arrangement. The upper fold of the outer belt should be supported in an idler, as shown at *A* in Figs. 1 and 2. Then for rotation of the pulleys in the direction indicated by the arrows, Fig. 1 or Fig. 2, the clearance between the belts can be regulated by adjusting the height of the idler or by changing the length of the outer belt. The idler pulley should be 24

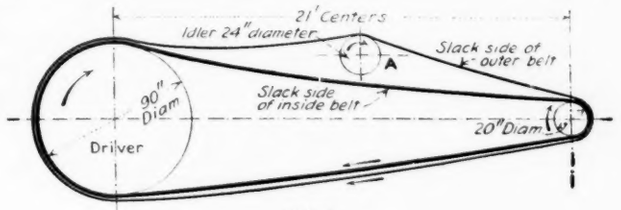


FIG. 1

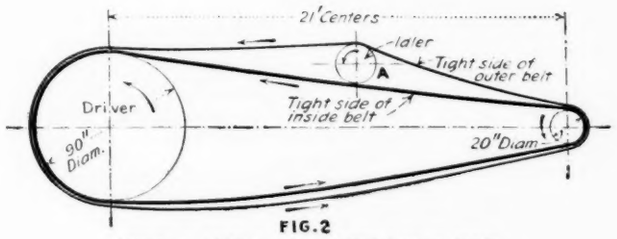


FIG. 2

FIGS. 1 AND 2—OUTSIDE BELT FOR INCREASING TRANSMISSION

in. in diameter or larger, have a slightly crowned face, and be mounted on a rotating countershaft. Evidently, the present belt is overloaded and by the foregoing arrangement the belt tensions may be less than at present, and if the load is not in excess of the proper combined capacity of the belts, deterioration should be no greater than in ordinary belt drives.

Advantages of Low-Pressure Steam Heating

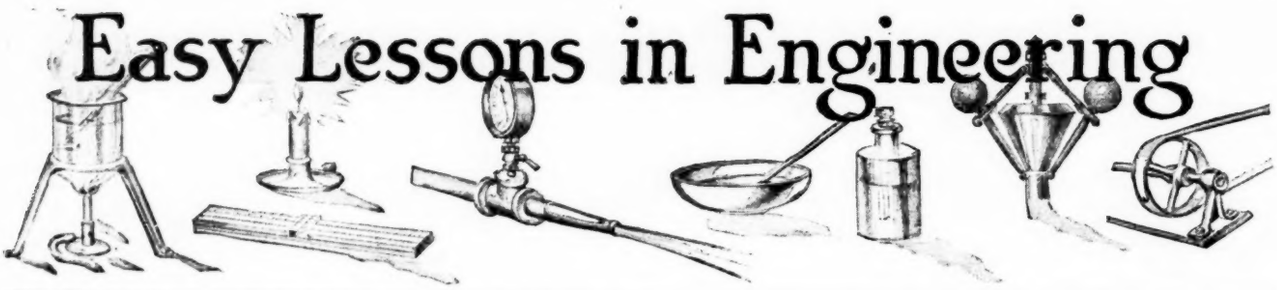
In heating with live steam is there any gain in the use of a reducing valve in place of obtaining heat directly from coils and radiators supplied with high-pressure steam?

F. P.

There is neither loss nor gain of heat from discharging steam through a pressure-reducing valve, but for warming a room with steam at low pressure there is better utilization of the heat supplied as a result of evener room temperature, less overheating near upper strata of air in the heated rooms and of spaces in the immediate vicinity of radiators; more complete absorption of heat from lower temperature of the returns; and less discrepancy of pressures in returns from different radiators or different sections of a heating plant, resulting in more economical operation, from less waste of heat returned from more active radiators to obtain circulation in others.

In any plant where high-pressure steam is used for pumping or other power, the low-pressure heating permits utilization of the heat of the exhaust, and for a large building there is an advantage of using steam reduced from a higher pressure, as there may be wide variation of the boiler pressure carried without affecting the lower pressure required to supply the heating apparatus.

Easy Lessons in Engineering



Strength of Riveted Joints

Fluid pressure inside the shell of a tank or boiler puts a tension on the riveted joints. These are always weaker than the body of the metal, so that the investigation of the strength of any "pressure vessel" consists largely in a study of the strength of the riveted joints.

Consider first a lap joint with a single row of rivets as shown in Fig. 1. For purposes of study, this joint

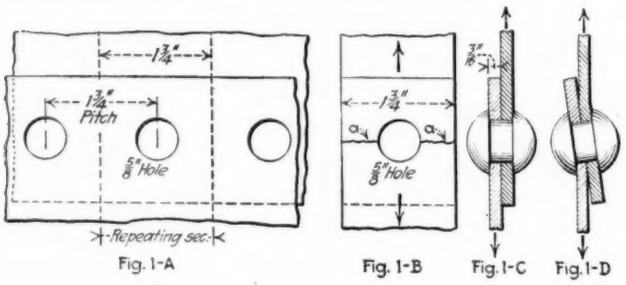


FIG. 1—SINGLE-RIVETED LAP JOINT

can be considered as broken up into a series of "repeating sections," as indicated in *A* and drawn separately in *B* and *C*. The section shown in *B* may fail in various ways. The plate may break in tension along a line *aa* passing through the hole. The rivet may shear off. The rivet may break through the edge of the plate. Finally, the rivet may crush the plate.

Evidently, a given joint will fail at its weakest point. The possibility of the rivet breaking through the edge of the plate need not ordinarily be considered, as failure in this manner is easily avoided by extending the plate a reasonable distance beyond the row of rivet holes. This leaves to be considered the breaking of the plate in tension along *aa*, the shearing of the rivet, and the crushing of the plate by the rivet. Suppose the plate material has an ultimate tensile strength of 55,000 lb. per sq.in., the rivet shearing strength of 44,000 lb. per sq.in., and the plate a crushing strength of 95,000 lb.[1]

The efficiency of a joint is the ratio of its strength to that of the solid plate back of the line of holes. The efficiency is computed separately for each kind of possible failure and the lowest of these taken as the efficiency of the joint. Taking up first the failure of the plate metal in tension, it is seen that a ⅝-in. hole in a section 1¾ in. wide removes 35.7 per cent of the metal, so that the remaining metal has 64.3 per cent of the strength of the undrilled plate. Hence the efficiency against the plate failing in tension along a line *aa* is 64.3 per cent.

[1]This would not be the actual compressive strength of the material of the plate, but a figure found experimentally to give the correct results if the area subject to compression is taken as the projected area of the hole; that is, the product of the plate thickness by the hole diameter.

Next comes the shear of the rivet. The shearing area is that of a ⅝-in. circle, or 0.306 sq.in. Allowing a shearing strength of 44,000 lb. per sq.in., the rivet would shear at a load of $0.306 \times 44{,}000 = 13{,}464$ lb.

The projected area of the rivet hole is $\frac{5}{8} \times \frac{3}{16} = 0.117$ sq.in., so the force required for the rivet to crush the plate would be $0.117 \times 95{,}000 = 11{,}115$ lb.

The tensile strength of the solid plate is $1\frac{3}{4} \times \frac{3}{16} \times 55{,}000 = 18{,}047$ lb. Therefore the efficiency of the joint against shearing the rivet is $13{,}464 \div 18{,}047 = 0.74$, or 74 per cent, and against crushing the plate is $11{,}115 \div 18{,}047 = 0.62$, or 62 per cent. Since the joint will fail where it is weakest (in this case by crushing the plate) the efficiency of the joint as a whole is 62 per cent.

A lap joint, such as this, has a serious defect in that it tends to pull out of line as shown at *D*. In boilers this may lead to "lap-seam cracks."

This danger is avoided by the use of a double-strap butt joint like that shown in Fig. 2. The joint shown is single-riveted. Higher efficiencies may be obtained by putting two or more rows of rivets on each side. A repeating section of this joint is shown at *B* and *C*. Being in series like the links of a chain, each half of this section sustains the full load on the section. Considering then the top half only, the efficiency against breaking the plate along a line through the hole is figured in the same way as for a lap joint. The same

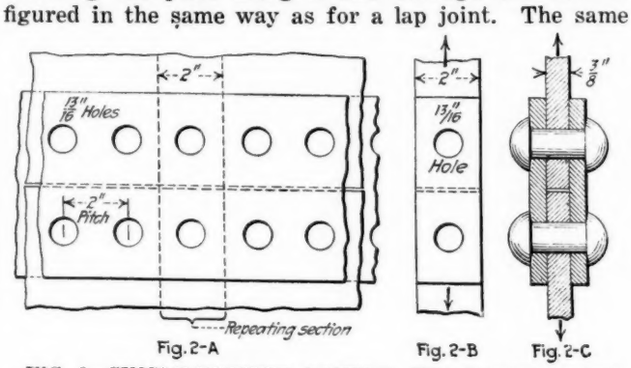


FIG. 2—SINGLE-RIVETED DOUBLE- STRAP BUTT JOINT

applies to the crushing of the plate by the rivet, since the full load on the section is exerted on the walls of the hole in the main plate. For the joint to fail by shearing the rivet, however, it would be necessary to shear it in two places, so that the shearing strength is based on twice the cross-sectional area of the rivet.

The design of riveted joints, particularly in boilers and other pressure vessels where safety is the first consideration, should never be attempted except by those thoroughly trained in all the theoretical and practical details of the work. The present article does not attempt to do more than give a general idea of how the design and dimensions of a riveted joint determine its efficiency.

Methods of Reducing Loss in Steam Exhausted From Turbines*

IN the design of steam turbines for large outputs at high rotational speeds, the limit of output is nearly always determined by the allowable magnitude of the leaving losses.

The weight of steam leaving the last row of blades of a turbine is given by

$$G \text{ (lb. per sec.)} = \frac{\tau \pi DhV}{144v}$$

where

D = Mean diameter in inches;
h = Blade height in inches;
V = Steam velocity leaving the last row of blades in an axial direction in ft.-sec.
v = Specific volume at turbine exhaust in cubic feet per pound;

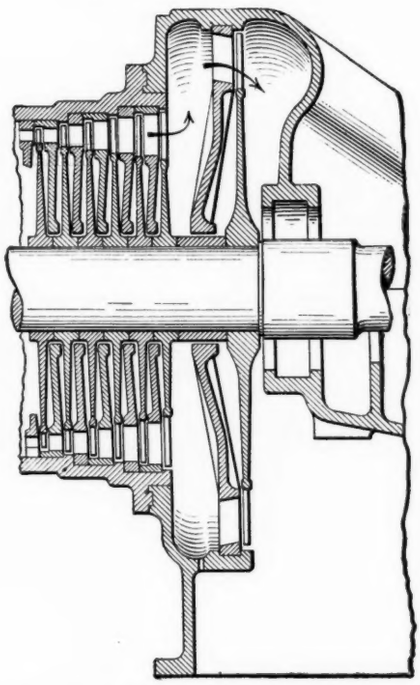


FIG. 1—LARGE DIAMETER AND RELATIVELY SHORT BLADE LENGTHS AT LAST DISK PRODUCE HIGH BLADE SPEED, PASSING A LARGE VOLUME EFFICIENTLY

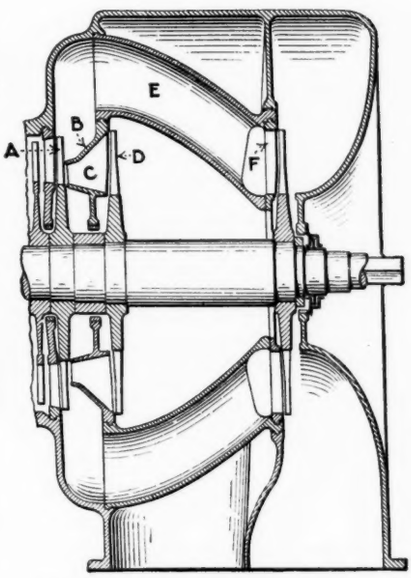


FIG. 2—DIAPHRAGM *B* DIVIDES STEAM FROM *A* INTO PATHS *C* AND *E*. THUS OBTAINING TWO ROWS OF EXHAUST BLADING, *D* AND *F*

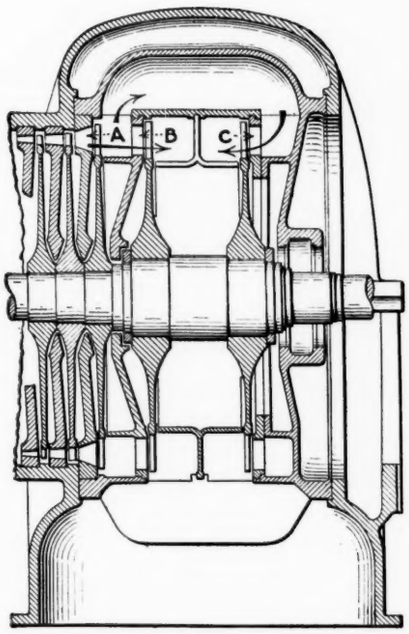


FIG. 3—DOUBLE FLOW FROM STAGE *A* INTO BOTH *B* AND *C* PROVIDES TWO EXHAUST BLADE ROWS *B* AND *C* EXHAUSTING TOGETHER

τ = Thickness co-efficient of last row of blades which may be taken as 0.95.

The output is given by

$$\text{KW. } (kilowatts) = \frac{\tau \pi DhV \times 3600}{144vS}$$

where S is the steam consumption in pounds per kilowatt hour.

Therefore, the output which can be obtained depends on:

1. The vacuum which fixes the specific volume.
2. The mean diameter, which for a given number of revolutions per minute depends on the allowable peripheral speed.
3. The blade height, which depends on the design and allowable stresses in the blades.
4. The leaving velocity, which is determined by the leaving loss that can be allowed.
5. The steam consumption, which is settled by the steam conditions and the turbine efficiency.

Where maximum efficiency is desired, as, for instance, in large turbines, the relation between the steam and blade speeds in the last stage is such that the steam velocity leaving the last wheel is in an axial direction or very approximately so. The leaving loss in B.t.u. per pound is then given by

$$L = \frac{V^2}{2gJ}$$

where

V = Steam velocity leaving the last row of blades in feet per second;
J = Mechanical equivalent of heat.

And if i = total available heat drop in B.t.u. per pound, the leaving loss expressed as a percentage of the total available heat drop is

$$\frac{V^2}{224^2 \times i} \times 100$$

Taking the steam conditions as being 250 lb. per sq.in. gage, 650 deg. F. total temperature, and 29 in. vacuum, we get the leaving loss in percentage for various leaving velocities as shown:

Leaving velocity
300 400 500 600 700 800 900 1000 ft. per sec.
Leaving loss—per cent
0.40 0.71 1.11 1.60 2.17 2.84 3.59 4.44

The leaving area of the last row of blades is

$$A = \tau \pi Dh$$

where τ = 0.95 and for comparison purposes may be omitted.

The leaving area can, therefore, be increased by increasing the mean diameter, the blade height, or both, the limit being the maximum permissible stresses in the blades and

*From an article by I. R. Cox "Devices to Reduce Leaving Losses in Steam Turbines," in *Beama* (London) September, 1923.

wheels, subject, of course, to their stiffness being satisfactory.

Considering now blades of uniform cross-section, the stress at the root is given by

$$\sigma = \tfrac{1}{2}\,\mu\,\omega^2\,Dh$$

Where μ = specific mass of material in appropriate units and ω = angular velocity in radians per second.

Put $A = \pi Dh$

$$\therefore\ \sigma = \frac{1}{2\pi}\,\mu\,\omega^2 A = Constant \times \omega^2 A$$

whence it will be noted that the blade stress for a given r.p.m. is proportional to the leaving area. Also for a given permissible blade stress the leaving loss varies directly as the fourth power of the r.p.m., that is, the problem of providing adequate leaving area becomes increasingly difficult at higher rotational speeds.

The first and obvious method of obtaining greater leaving area is by increasing the mean diameter of the last stage without decreasing the height of the last row of blades. We have shown that for a given number of revolutions per minute the stress in the blades is proportional to the blade area, so that the adoption of this method leads to higher blade stresses, involving, if the factor of safety is kept normal, the use of materials that have unusually high physical properties. As a general proposition it may be stated that it is not advisable to utilize materials of special and unusual physical properties in order to maintain factors of safety. For the high-pressure end of the turbine, disks or a drum may be used, but it becomes necessary in both impulse and reaction machines of large outputs to use disks for the last stages, so that this method involves the use of large disks running at a high peripheral speed. Such disks are, of necessity, highly stressed and difficult to design if they are to be immune from vibration troubles.

The frequency of vibration of a disk depends, among other things, on the speed, its shape and dimensions, the pull of the blade it carries and on the forging stresses. In addition, it is affected by a difference in temperature in various parts of the disk. The effect of only some of these is calculable; all become more difficult to guard against as the disk diameter is increased. The modern tendency to keep disk diameters down to a moderate size is, without doubt, wise and good design. The experience of turbine builders in the United States with disks of large diameters has been of great service to turbine designers in this country in impressing the necessity for conservatism as regards wheel diameters and peripheral speeds. Mean diameter speeds of 750 to 800 ft. per sec. were at one time common in America, and trouble due to high centrifugal stress and vibration was experienced.

The present state of good design is that 650 ft. per sec. mean peripheral speed is the figure beyond which it is unwise to venture. As regards material and permissible stresses standard materials of 40 to 48 tons per square inch ultimate tensile strength for both wheels and blades are readily obtainable. It would represent safe and conservative practice to take the lower figure and aim for a factor of safety of five on the ultimate tensile strength.

In order to increase the leaving area without increasing the blade stress, it is necessary to depart from the parallel blade and adopt a blade the section of which is larger at the root than at the tip. The use of taper blades can be said to constitute a real device for reducing leaving losses. It is assumed that the blade can be fastened to the wheel so that the stress in the fastening is not greater than the stress in the blade. With relatively long blades this requires careful designing, but it is certainly possible. A blade may be tapered in two ways; it may be thickened toward the root or widened. The pitch of the blades at the tip must be such as to insure adequate guiding of the steam, and only such thickening at the root is permissible that the steam is not throttled in passing through the blades; this imposes a limit as to thickening the blades. Widening the blades toward the root can also be done, provided it is not carried so far that the attachment to the wheel and the design of the preceding diaphragm become impossible.

By suitable thickening and widening of the blades and without increasing stresses, it is possible, consistent with good design, to increase the leaving area by approximately 100 per cent, about one-half of this being due to thickening and one-half to widening. This device properly applied, therefore, can be utilized to cause a reduction of 75 per cent in the leaving loss for a given stress and a definite r.p.m. It is perhaps instructive to compare two different methods of obtaining a given leaving area: First, large diameter disk—relatively short blade. Second, small diameter disk—relatively long blade.

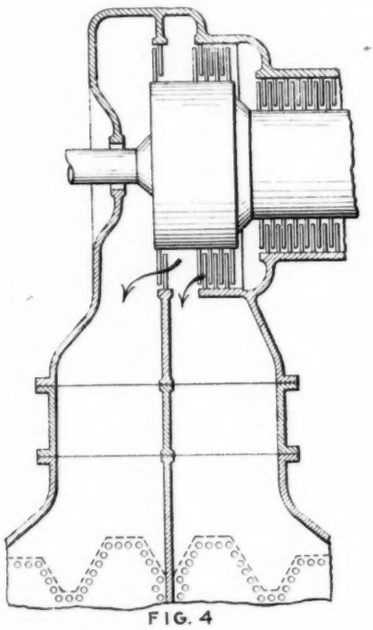


FIG. 4

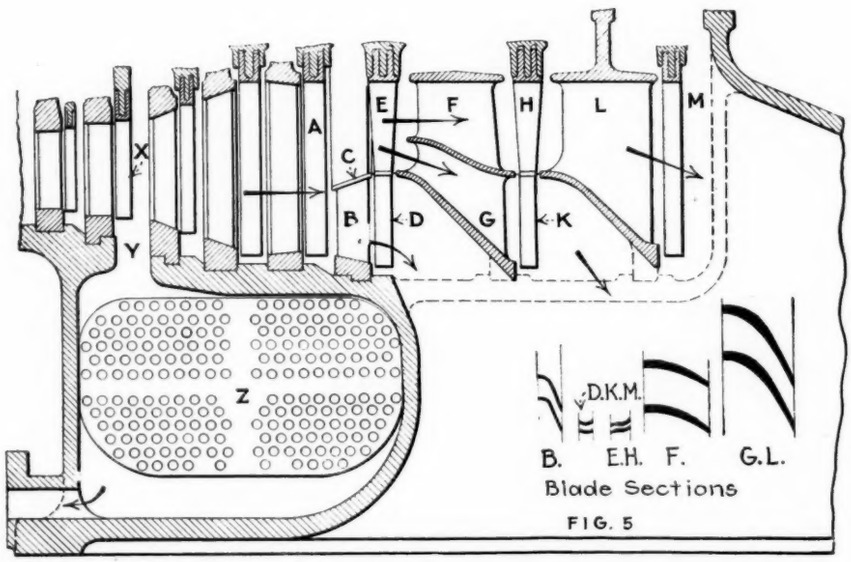


FIG. 5

FIGS. 4 AND 5—SHOW LEAVING LOSSES DECREASED BY DIVIDED CONDENSER AND DIVIDED STEAM PATH

Fig. 4—Part of exhaust steam enters condensers at right. Remainder passes final blade row to lower vacuum of condenser at left. The object is to obtain increased output with a given quantity and inlet temperature of circulating water, which flows through the higher vacuum condenser first and then through the lower.

Fig. 5—Bauman principle applied to impulse turbine secures increased capacity with moderate stresses and leaving losses. Three exhaust rows, *D*, *K* and *M*, are here provided by dividing the path of steam expansion, beginning at *A*. Three exhaust areas, *D*, *K*, and *M* are thus obtained.

For equal leaving areas the blade stresses will be the same; the larger disk will have the higher peripheral speed and will, therefore, be more highly stressed. A small disk can be made stiff enough to preclude any possibility of synchronous vibration, whereas it is difficult to design a large disk which is safe in this respect. The long blade will, of course, need to be secure from vibration, but this is more easy to arrange for than security from vibration of a large diameter disk.

The larger variation in blade pitch with the longer blade adversely affects the efficiency slightly, but this is outweighed by the increased safety and less tendency to vibration troubles. It will be appreciated that in dealing with blade stresses, not only must those due to centrifugal force

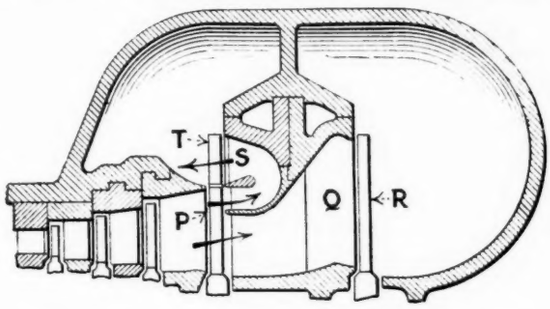


FIG. 6—DIVIDED AND REVERSED STEAM PATH INCREASES EXHAUST BLADE AREA

Steam at *P* enters the two paths *Q* and *S*. The former exhausted directly through blade *R*, and the latter, after reversing, through tip section *T* of same blade row *P*.

be considered, but care must be taken that the dimensions of the blades are such as to maintain at a small figure, compared with the centrifugal stresses, the stress due to bending caused by steam action. This stress is usually not more than one-fourth of that due to centrifugal force.

Special Designs Reduce Leaving Losses

The devices utilized for decreasing the leaving loss, or for increasing capacity, are shown in Figs. 1 to 6. The relatively large disk with short blade lengths in Fig. 1 involves a lower factor of safety with regard to both stress and stiffness for this stage than for the preceding.

In Figs. 2 and 3 two rows of exhaust blading per rotor are employed. Each plan involves the disadvantages of friction loss due to an external steam passage.

In Fig. 4 part of the exhaust steam is condensed at a somewhat higher pressure than the remainder, which is discharged through the last blade row and condensed in a separate compartment of the same condenser. At Fig. 5 is shown Bauman blading as applied to an impulse turbine. A description of the Bauman principle appeared in the May 15 issue of *Power*. A heater is employed at *Z*, for heating feed water by the steam at the point *Y*. Steam here is exhausted from the blading at *X*. Condensation is drained at the bottom. The Bauman principle allows low leaving losses without the friction incident to external steam passages. In Fig. 6 the steam flow is divided. One path *S* reverses the direction of flow and utilizes a row of blades *T* superimposed on the one from which the steam issued, *P*.

Is smoke harmful? Why do people go to the country? Partly on account of change of scenery of course, but partly also on account of change of air. Why is not poetry written about smoke, fog and dirt as it is about pure air and sunshine? Does any one want to claim that pure-water and pure-food laws are not essential and that they should not be enforced, that it will cost the taxpayers too much because the city purifies the water, that it is too inconvenient and too expensive for the canners and food caterers, because they must preserve and serve pure foodstuffs. If there is no question about necessity for pure food or pure water, there should not be any question about the necessity for pure air.—From an article by V. J. Azbe in *Forge*.

Irregularity of Reaction in Francis Turbines*

By Roy Wilkins†

During the last few months there has arisen an occasion to study the hydraulic conditions causing vibration in some of the Francis-type turbines on the system of the company by which the writer is employed. Preliminary reconnaissance showed that these vibrations had a period in the order of the electrical frequencies commonly used and that for their study special equipment must be used which, so far as is known, had not been developed. There was accordingly developed somewhat hurriedly from parts available, a device for converting pressure variations to electrical impulses and then viewing or recording these by means of an oscillograph.

As shown in Fig. 1, a 1½-in. pipe union was used to hold a diaphragm of spring steel of a thickness such that at the maximum pressures used its deflection is still approximately proportional to pressure change. Several well-known gas-engine indicators operate on this principle, and it is known to be trustworthy. Rigidly attached to its center is an arm bearing on its outer end the center elements of two microphones of a variety used commonly in amateur wireless work. The outer elements, or shells, of these microphones are rigidly attached by an insulating member to the union in such a manner that the supports are used as current-carrying terminals. A third terminal grounded to the union makes electrical connection through the arm to the center elements of the microphones.

With the connections as shown the device forms an elementary bridge with microphones and batteries as members and an oscillograph element as a detector. The 2-volt bat-

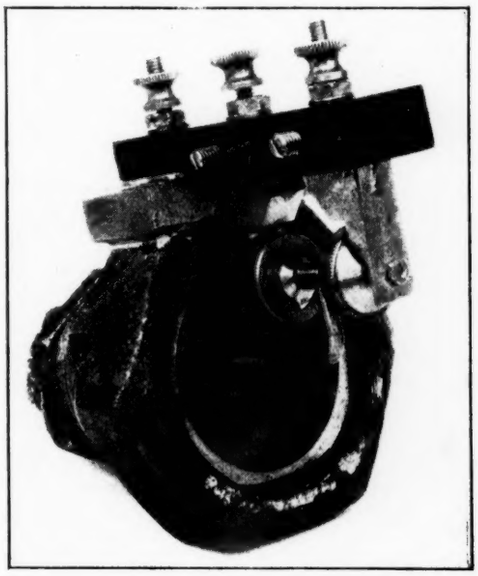


FIG. 1—DEVICE USED FOR CONVERTING PRESSURE VARIATIONS INTO ELECTRICAL IMPULSES

teries forming two arms of the bridge are so connected that they tend to cause current to flow through the oscillograph element in opposite directions so that as the diaphragm moves owing to pressure changes, a curve is produced which resembles an alternating-current wave traced on the oscillograph film, proportional to and in correct sequence with the pressure changes back of the diaphragm. The device can be attached to any point that an ordinary

†Assistant engineer, Pacific Gas & Electric Company.

*Abstract of paper presented at the Pacific Coast Convention of the American Institute of Electrical Engineers, Del Monte, Calif., Oct. 2 to 5, 1923.

pressure gage can, and a ½-in. connection, if comparatively short, is more than ample to actuate it.

Using one of the portable oscillographs now available, studies have been made on several turbines under varying conditions of load, head, gate opening, etc., with a view of determining the fundamental reason for vibration. It is known that the pressures across the face of a turbine-runner opening vary both from top to bottom and from trailing to leading sides and are dependent upon design characteristics as applied to running conditions. As a check on probable values, tests were made, using a Pitot tube

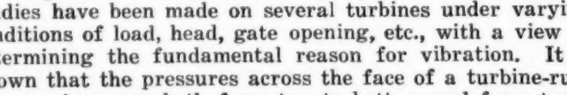


FIG. 2—DIAGRAM OF THE DEVICE FIG. 1 CONNECTED TO AN OSCILLOGRAPH

traveling with the runner, Fig. 3, and passing to the center of the draft tube and out through a packing gland either to a pressure gage or the pressure indicator as previously described. By filling this rotating tube with air under pressure and then closing, the supply air is trapped under the pressure of the water at the end of the tube and centrifugal action and leakage are minimized. In this way the whole face of the runner entrance may be explored under actual running conditions.

Having established the fact that there is a difference in pressure on the water over the entrance of a runner bucket, a clue is given to the reason for the changes in pressure on the scroll case and penstock supplying it, because as a low-pressure area in the runner is presented to the guide-vane opening, the water velocity is augmented, while when

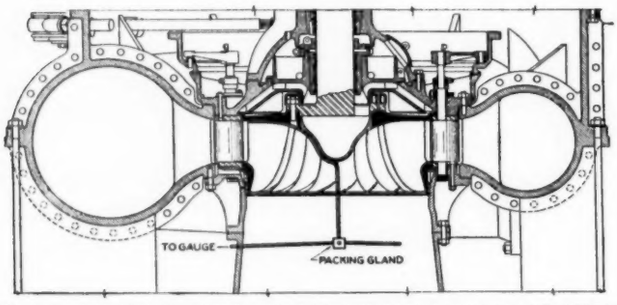


FIG. 3—SECTION THROUGH TURBINE SHOWING LOCATION OF PRESSURE PIPE

a high-pressure area is presented the entering velocity is checked.

This difference in pressure has been measured where the difference between leading and trailing sides of the runner-entrance pressures was approximately 10 per cent of the total head on the turbine.

With a turbine in which the number of runner-vane openings ($N - 1$) is one less than the number of guide-vane openings, N, the openings will register successively around the scroll case each $\frac{1}{N-1}$ revolution as a vernier backward from the direction of rotation. This gives ($N - 1$) impulses per revolution, or the revolutions per second times the number of runner vanes in impulses per second.

A difference of two in the number of guide and runner vanes gives two impulses traveling at half-speed around the scroll case, while combinations such as 20 and 17 give irregular impulses. These combinations are best studied on small drawings to scale on cardboard, the runner portion of which may be rotated by hand. Fig. 4 shows a pressure curve of a turbine having 20 guide vanes and 19 runner vanes with a timing wave of 60 cycles. This gives the vernier action described in the foregoing, and the machine has a decided hum and vibration which persists with gradually lessening intensity up the penstock 3,600 ft. to the forebay, where it is scarcely perceptible.

In taking such records, the so-called auxiliary data are of as much importance as the record itself, and all conditions should be faithfully recorded in order to study and fully appreciate all the changes made evident by a film of pressure variations.

Only ordinary precautions are necessary in using the apparatus, such for instance as making sure that there is no trapped air behind the diaphragm, that all electrical connections are good and that the oscillograph is properly adjusted and handled.

The microphones are adjusted originally by disconnecting one microphone lead and observing the shift in the zero line caused by current through the one connected and then repeated with the other, adjusting until the shift is the same for each. The adjustment in the present device consists in moving the microphone shell in or out a small distance with respect to the central part fast to the diaphragm.

It is but a few minutes' work to set up and observe the waves, and changes in pressure waves may be watched

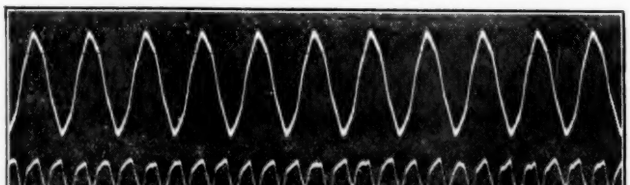


FIG. 4—PRESSURE CURVE OF TURBINE HAVING 20 GUIDE VANES AND 19 RUNNER VANES, WITH A TIMING WAVE OF 60 CYCLES

Load on unit 5,000 kw.; speed, 450 r.p.m. There was a hum apparent in both the power house and penstock.

while operating conditions are changed.. Once adjusted, different points may be explored wherever a half-inch pipe connection can be made.

Nearly all the cases studied have to do with the manner in which the water enters the runner and are fundamental in turbine design.

It is thought that the shock due to sudden changes in velocity of moving water has not been fully appreciated in many cases.

Using the ordinary formulas for surges and water hammer as given in Prof. Durand's "Hydraulics of Pipe Lines," it is seen that with 80 cycles per second, that is, 160 changes, there is required somewhat less than four one-hundredths of a foot change in velocity per second to cause 100 lb. pressure change, provided all this change were effective.

Elasticity in the scroll case and penstock and large clearances between guide and runner vanes together with the combinations of guide and runner vanes which set up conflicting pressures, all tend to prevent the difference in pressure across a runner opening from causing high-pressure variations in the scroll case and penstock.

Certain design characteristics cause most of the energy in the water entering the runner to be in the form of velocity and therefore incapable of causing any of the phenomena observed. In other designs more reaction and therefore more pressure is necessary, and greater care must be used in entrance angles and clearances.

The subject is as yet very new and very little concerning it is known. It is only hoped that such work as has been done may be of assistance in further study.

Problems on Electric Drive for Ventilating Equipment

At the regular monthly meeting of the New York Section of the American Institute of Electrical Engineers the problems involved in the application of fans and electric drives to fans were presented in four papers. In presenting his paper, "Direct-Connected Motor Driven Fans," F. R. Still said that how to attach alternating-current motors to fans and blowers that are to be used for heating and ventilating buildings is a problem that almost every engineer spends more time uselessly trying to solve than is spent on any other engineering problem he encounters. The speaker pointed out that, on account of the low speeds at which heating and ventilating fans had to operate to be silent, the problem of connecting them directly to alternating-current motors is a difficult one. It has been determined by careful tests that the so called high-speed fans cannot operate at tip speed exceeding 6,500 ft. per min. and be silent, and it is not always safe to operate them in some buildings at this speed. This makes it necessary to operate these fans at slow speeds not suited to alternating-current motors. When fans of this type are used for forced draft in connection with stokers under boilers, noise is usually not a serious factor for consideration. As the pressures required for such work vary from 4 to 7 in., usually ranging between 5 and 6 in., and the tip speed frequently is from 12,000 to 15,000 ft. per min. for such duty, the "high-speed fan" lends itself to direct-connected alternating-current motors.

It was the speaker's opinion that where alternating-current motors have to be used for driving fans in public buildings, wherein quietness is essential, it is advisable to install a slow-speed fan and drive it by a belt from an alternating-current motor. Where direct current is available, combinations that will permit of using direct-connected motors can easily be arrived at.

C. A. Booth, in his paper, "Direct Motor Drive for Ventilating Fans," agreed with Mr. Still concerning the use of belts for driving fans that had to operate noiselessly, and said that as usually applied for commercial work, there is no space or other limitation that is likely to prohibit the use of belt drive for fans. The speeds and horsepowers are moderate, the ordinary size of unit requiring from 3 to 10 hp., the fan speeds are such as to give a pulley ratio averaging about 4 to 1, and the use of standard equipment appeals strongly to the manufacturer who may have a number of duplicate motors throughout his plant. Belt drive is by far the most common arrangement for heating, ventilating and drying fans in factories.

For forced- and induced-draft work the fans usually operate at somewhat higher pressures, and therefore at higher speeds, besides which space limitations frequently prevent the use of belts, and when electric drive is selected, it is frequently necessary to couple the motor to the fan shaft.

After reviewing the different developments in fan design for ventilation and forced-draft purposes, the speaker described a high-speed type the essential features of which are: Greater radial depth of blade in proportion to the diameter of wheel; a double-curvature blade in which the inner edge, or heel, picks up the air without shock, and the outer or tip edge is turned backward away from the direction of rotation; an inlet cone on the housing similar to the construction used in hydraulic turbines for gradually increasing the velocity of flow.

To the motor manufacturer the advantage of the high-speed fan is apparent, and further developments along this line are greatly to be desired, but even with the present speeds it becomes possible to use units which are more efficient by reason of their higher speed, require less space and less expensive foundations, and it also makes the more general use of direct motor drive possible by reducing the excessive cost and the time of delivery required for special slow-speed machines. It would be greatly to the interest of the public to have closer co-ordination between motor manufacturers and the fan companies, so as to advise engineers and architects of the possibilities in this direction.

C. F. Scott presented a paper on "The Application of Electric Motors to Blowers," and said in part: Direct-current motors for blowers employed in ventilating systems service should be shunt wound. It is comparatively easy to obtain almost any speed required on a slow-speed direct-current shunt-wound motor, but there are certain distinct features that are of importance to the designer and to the user, but are often not fully appreciated by the architect or engineer under whose supervision the building is to be constructed or equipped.

When it comes to alternating-current motors for the same service, the problem of connecting the motor is often difficult. Not only are small alternating-current motors for 60-cycle circuits expensive when built with the large number of poles required for a synchronous speed of less than 600 r.p.m., but the electric characteristics, efficiency and power factor are quite poor. In the case of motors of less than 3 hp. one solution lies in the use of repulsion motors, to be connected single phase on the different legs of a three-phase or two-phase circuit.

While considerations of space, as for example, in the case of a fan mounted in a monitor, will sometimes absolutely preclude a belt or chain drive, nevertheless the belt or chain drive can be used in a large number of cases with alternating-current motors driving slow-speed fans.

Troubles experienced with belt drives can be frequently eliminated or largely reduced by the proper selection of belts, and if the same attention were given to the selection of the belt, its width, thickness, number of plies, material, method of making endless, speed in feet per minute, etc., as is given to other features of the fan installation, belt drives would undoubtedly meet with more general approval.

Silent chain drive has been used in a number of cases and might have a considerably wider application. By a careful selection of the right chain speed and the proper pitch, angle and tightness, the chain can be made quiet and the noise can be still further diminished where necessary by incasing the chain and running it in oil.

"Electrical Motors for Ventilating Fans" was the title of a paper presented by J. L. McK. Yardley, in which he called attention to some of the problems in applying motors to fans. The developments in recent years in fan manufacture, said Mr. Yardley, particularly in regard to increasing normal speeds and reducing the tendency to overload or underload, under resistance conditions different from normal, have materially increased the number of fan applications that may now be directly driven by standard motors. To meet all the conclusions of ventilating engineers, as to horsepower and speed required, has necessitated ratings of motors running up into the thousands. One electrical manufacturer, for example, for fan and blower service, has listed in his price form no less than 3,000 different ratings of 230-volt direct-current motors alone, in sizes from and including ½ hp. to 50 hp., and for motor speeds extending from 90 to 1,050 r.p.m., with speed-variation capabilities varying from 1¼ to 1, to 2 to 1 of the respective minimum speeds.

It is indeed difficult to see how we are to get away from having a fairly large number of motor ratings for ventilating-fan service and particularly how we are to get away from the very general employment of direct-connected variable-speed direct-current motor drive for ventilating fans in schools and public buildings, since they are satisfactory in every way except cost. Personally, I have some hope that this may come about in the future through some type of gear which, while being practically noiseless, will be sufficiently inexpensive to be applicable, in connection with standard squirrel-cage induction motors.

Discussion of the Papers

In the discussion the necessity for closer co-operation between the fan and the motor manufacturers, toward using motors of standard ratings and speed, was emphasized. A greater use of such motors would not only help to keep down the cost, but also simplify the problem for the user in providing renewal parts. It was pointed out that the crane manufacturers had greatly simplified their motor problems as well as those of the users by standardization.

News in the Field of Power

Appropriation for Study of Engineering Education

The Carnegie Corporation has recently made an appropriation of $108,000 for the study of engineering education. W. E. Wickenden, assistant vice-president of the American Telephone & Telegraph Co., has been appointed director of the work. He is to work in co-operation with the Society for the Promotion of Engineering Education, make affiliations within the engineering schools of this country, and also visit the engineering schools of Europe.

Heating & Ventilating Society Nominate New Officers

The Nominating Committee of American Association of Heating and Ventilating Engineers is submitting the names of new officers for 1924 to its membership as follows:

President, Homer Addams, New York City; first vice-president, S. E. Dibble, Pittsburgh, Pa.; second vice-president, William H. Driscoll, Long Island City, N. Y.; treasurer, Perry West, New York City; members of council, F. Paul Anderson, Pittsburgh, Pa., Willis H. Carrier, Newark, N. J., Joseph A. Cutler, Chicago, Ill., Walter E. Gillham, Kansas City, Mo., L. A. Harding, Buffalo, N. Y., Alfred S. Kellogg, Boston, Mass., Thornton Lewis, Philadelphia, Pa.

Priest Rapids Project Ready to Start

A significant feature of the application of the Washington Irrigation and Development Co. for a license covering its Priest Rapids development is the statement in the application that construction work is to begin immediately upon the issuance of the license. The first block of 200,000 hp. is to be ready for delivery in four years.

Priest Rapids is on the Columbia River, halfway between Spokane and Portland. The Washington Irrigation and Development Co. proposes to build a dam 2½ miles long. The shore ends of the dam are to be of the earthen-wing type. In between will be the power-house section, which in itself will be 660 ft. long. It will provide for a 700-ft. future extension. There is to be a concrete spillway, with a leg extending along an island for five-sixths of a mile. In the spillway sector, there are to be 127 gates, each 30 ft. long, with a capacity sufficient to discharge a flood of 1,200,000 cu.ft.-sec., or more than five times the flow of the Niagara River. The largest flood of record at this point carried 710,000 cu.ft.-sec. Thus a large factor of safety is provided.

A head of 70-ft. will be developed at this dam. Provision is made for two navigation locks and for a fishway. The initial installation will provide six 40,000-kva. and two 29,000-kva. turbo-generators. The ultimate installation will provide ten additional 40,000-kva. units, thus providing for the generation of 748,000-hp.

Deerfield River Project Progressing

At Davis Bridge on the Deerfield River in the southern part of Vermont the New England Power Co. is building a hydro-electric plant that will ultimately have an installed capacity of 60,000 hp. The present installation will be two 20,000-hp. machines. The dam is the highest earth dam in the world, being 200 ft. high with a length of 1,250 ft. and a width of 1,300 ft. at the base, and contains 1,900,000 cu.yd. of material. This dam will form a reservoir with an area of 3½ sq.mi. and will contain 115,000 acre-feet of water, with a maximum of 185 ft. A spillway of unique design is used, which is circular in shape and resembles an inverted phonograph horn, being 160 ft. in diameter at the mouth, which gives a spillway length of 500 ft. Water discharges over the edge of this spillway, drops vertically 180 ft. and discharges through a concrete tunnel 22½ ft. in diameter. This spillway has a discharge capacity of 30,000 cu.ft.-sec. A concrete-lined pressure tunnel 14 ft. in diameter runs in a straight line from the dam to the power house, a distance of 2½ miles, and will have a normal capacity of 1,400 cu.ft.-sec. A differential surge tank is to be used, and on account of drawing the reservoir down 90 ft. for storage, the surge tank is of exceptional height, 184 ft. by 34 ft. diameter.

A maximum head of 390 ft. and a minimum of 300 ft., with a mean of 350 ft., will be obtained on the turbines, which are of the vertical single-runner type to operate at 360 r.p.m.

At the present time the dam is nearing completion; the spillway excavation is completed and the concrete for the spillway control is about to be placed; the intake tower will be finished in about a month; the power tunnel is holed through; the penstocks and surge tank are being erected; the power house is being bricked in and the work is going forward rapidly to completion. The dam has been rushed to completion considerably ahead of the power house in order to store the spring runoff for use next season. The Power Construction Co., Worcester, Mass., are the engineers and contractors, and the J. G. White Engineering Corp., New York City, are the consulting engineers.

Tentative Program for the N. A. P. R. E. Announced

The National Association of Practical & Refrigerating Engineers has sent out the tentative program for its Fourteenth Annual Convention and Exhibition which is to be held in Memphis, Tenn., Dec. 12-15. Papers of Interest to *Power* readers are as follows: "Open Brine Refrigerating Systems," by S. C. Bloom; "High-Speed Compressor Valves," by J. H. H. Voss; "Internal-Combustion Engines in Ice and Refrigerating Plants," by A. E. Park; "Evaporating Systems for Refrigerants," by W. F. Davis; "Railway Refrigerator Car Icing Systems," by A. L. Blatti; "Amount of Ammonia Required for Initial Charges," by J. Laichinger; "Cooling Water for Ice Making and Refrigerating Plants," by J. P. Lawlor; "Control of Electrical Equipment in Ice Making and Refrigerating Plants," by H. P. Hill; "Some Causes and Remedies for Cracked and Checked Ice," by Erle Ormsby; "Improvements of Ice Plants," by R. C. Doremus; "The Operation of the Absorption of Refrigerating Plants," by G. A. Pardee; "Cold-Storage Refrigeration," by Charles W. Chapman; "Raw-Water Ice-Making Systems," by Van Rensselaer H. Greene; "Safety Movements and Devices in the Refrigerating Industry," by M. F. Nicholson; "Relation of Education and Engineering," by Harry Sloan; "Latest Data on Some Common Refrigerants," by William H. Molz; "Burning of Oil Under Power Boilers," by R. F. Burke; "New and Suggested Forms of Ammonia Condensers," by Thomas Shipley.

An address will be given by R. H. Switzler, president of the A. A. I. R., on "American Association of Ice and Refrigeration." Motion pictures: "The Romance of Ice," by the N. A. I. I.; "Preservation of Perishable Foods," by A. A. I. R.; "The Keokuk Dam." Topics for discussion include: "Methods of Purging of Refrigerating Systems"; "Ice Storage-House Construction"; "Some Causes and Remedies for Exceptionally High Condenser Pressure"; "New and Improved Apparatus, Insulation, etc."; "Testing of Ammonia, Brines, etc."; "The Domestic Refrigerating Machine"; "New Methods of Treating Water for Raw-Water Ice Manufacture"; "New Uses for Refrigeration"; "Amount of 'Frosting-Back' for Compressors"; "Material for Brines in Ice and Refrigeration Plants."

Refractories for Specifications Being Established

In co-operation with an advisory committee composed of producers and consumers of refractories, the Bureau of Standards has undertaken the establishment of specifications to be used by the government departments in the purchase of these materials.

An important part of this work, which is now nearly completed, deals with refractory brick for the lining of stoker-fired boiler settings. Brick of many representative brands were subjected to tests suggested by the advisory committee, and the results of these laboratory tests were then correlated with service records on brick of the same brands, gathered from various power plants.

The brick tested represented 42 brands supplied by users in all the important industrial districts of the country. Specimens have been subjected to the standard and modified load tests, and the endurance, constant volume and quenching tests. Absorption tests before and after various heat treatments were carried out, and the fusion points and chemical composition were also determined.

The results of the load tests checked closely those obtained in previous investigations and show clearly the relative behavior of brick from various districts in the United States. Results of the investigation also indicate a close relation between chemical composition, fusion point and resistance to sudden temperature change, and these also agree quite satisfactorily with the results of the endurance and constant volume tests.

The laboratory work has been completed, the necessary service data obtained and the results correlated. Proposed specifications were drawn up during the past month and discussed Oct 23 at a conference composed of the Federal Specifications Board's subcommittee and the advisory committee on refractories. It is evident that a most important advance step has been taken in preparing a set of specifications based on the results of service tests rather than merely the quality of the material.

Merrill Says Program of Superpower System Wider Than Any State Authority

Co-operation and Uniformity of Legislation by the States, and Administration by Federal Power Commission

IN HIS report to Congress O. C. Merrill, executive secretary of the Federal Power Commission, will say:

Since water powers will form an essential part of any comprehensive superpower system, the controversies concerning them require early solution in order that the way may be cleared for carrying out the general program. This program will be much wider than the territory or the authority of any individual state. State interests, nevertheless, must be harmonized in policy and program, which will be to the common interests of them all. This will require co-operative action and reasonable uniformity of legislation. There must be no state barriers against the interchange of energy. There must be no type of development that cannot become an integral operating part of the combined system. Legislation which interferes with the program should be repealed or modified; necessary affirmative legislation should be had; public officials of both state and nation should lend the program their support and the industry itself should harmonize its own conflicting interests. It should no longer be permissible for any utility to draw plans for future extension except in such manner that interconnections may be effected readily whenever its territory merges with that of any other utility.

Mr. Merrill also recommends that the water power act be amended so as to place the administration of all projects whether authorized by this law or prior to its enactment in the Federal Power Commission. He also suggests that the act be amended so that the amounts collected for the purpose of reimbursing costs of administration, be placed in a special fund to be devoted exclusively to the purpose for which collected and to be expended under the direction of the Commission.

Merger Between American and Canadian General Electric

The American General Electric Co. has purchased control of the Canadian General Electric. President A. E. Dyment, of the Canadian company, stated that such a community of ownership would result in much more intimate and effective co-operation between the two organizations than could be obtained under the present contractual relationship. The personnel of the staff will remain as at present.

Big Plant for Indianapolis Light & Heat Co.

Construction of another large electric plant to serve the Indianapolis and central Indiana territory will be completed late in 1925 by the Indianapolis Light & Heat Co., according to a recent announcement by C. C. Perry, president of the company. He said that preliminary work on the project has been begun on a 145-acre tract about eight miles southwest of the city on White River. The plant will cost about $5,000,000 and will involve an expansion program in addition to improvements now being made in the equipment of the company's physical property. These improvements, constituting the 1923 development program, are costing about $1,700,000.

Electricity Produced Daily for Nine Months

The average daily production of electricity for each of the first nine months of 1923 and the proportion produced by water power were as follows: January, 153,300,000 kw.-hr., 34 per cent; February, 154,400,000 kw.-hr., 33.9 per cent; March, 152,500,000 kw.-hr., 36.3 per cent; April, 149,100,000 kw.-hr., 39.9 per cent; May, 150,100,000 kw.-hr., 41.3 per cent; June, 150,800,000 kw.-hr., 38.9 per cent; July, 146,300,000 kw.-hr., 36.4 per cent; August, 150,800,000 kw.-hr., 33.8 per cent; September, 152,400,000 kw.-hr., 32.6 per cent.

New Jersey Desires To Sue the Federal Power Commission

New Jersey presented a motion for leave to file an original suit against the Federal Power Commission to the United States Supreme Court on Nov. 12.

The principal complaint of New Jersey is that the Federal Power Commission is attempting to exercise jurisdiction over the state's proposed development of water power along the Morris Canal, which runs from the Delaware River at Phillipsburg to the Hudson River near Jersey City, and on which New Jersey plans to develop power at various sites. It is said that the Commission is also asserting authority over riparian lands which bring revenue to the state for its school fund. Control by the Federal Commission also would interfere with the conservation policy of the state over its potable waters and cause heavy loss to investments in water-works, the bill declares. The authority of the Commission over certain streams and to regulate rates for the sale of electric power within the state is challenged.

Power Stations for the British Empire Exhibition

Work has now been started on the power station for the supply of electric energy to the British Empire Exhibition to be held at Wembley Park next year. The power station, which itself forms one of the exhibits in the Electrical and Allied Industries section of the Palace of Engineering, will include three 1,500-kw. turbo-generator sets supplied with steam from watertube boilers of two different makes. Each of the turbo-generators will be constructed by a different firm and the switchgear, condensing plant and auxiliary machinery generally will also be the work of several different firms. In addition a steam-driven dynamo and another driven by a horizontal internal-combustion engine, each of 300 kw., will work in the stands of the makers in the main hall and will be connected up to panels on the power-station switchboard. The whole of the electrical and allied engineering section of the exhibition has been organized by the British Electrical and Allied Manufacturers' Association.

Bloemfontein Power House Report Available

In connection with the new electric power station to be erected by the municipality of Bloemfontein, South Africa (see *Commerce Reports*, Oct. 29, p. 291), there are available for loan to firms interested two copies of a report on the power-house extensions made by the consulting engineers, whose recommendations have been adopted. Inquiry should be made direct to the Electrical Equipment Division, Bureau of Foreign and Domestic Commerce, Washington, D. C., mentioning file No. 108909.

New Publications

Mechanics of the Gasoline Engine. By H. A. Huebotler. Published by McGraw-Hill Book Co., Inc., New York. Cloth; 6x9 in.; 304 pages; 153 illustrations. Price, $4.

The automotive industry has grown so rapidly that engine designers have been hard pressed to outline the developments made. As a consequence the design of a new automotive engine has too frequently been a mere matter of copying parts of proven engines. H. A. Huebotler set for himself the task of outlining the principles of design, incorporating the general theories the young engineer or student must understand before seriously undertaking drawing-board work. While a mere study of this volume will by no means make one an engine designer, nevertheless it should serve most admirably as a general reference work and especially as a textbook for automotive engineering students.

Steam, Its Generation and Use. Thirty-sixth edition, 1923. Published by the Babcock and Wilcox Company, New York. Cloth; 7x10 in.; 383 pages; profusely illustrated.

The blue cloth binding with title and decorations in gold leaf cannot fail to draw attention. Photographs, drawings, halftone sectional illustrations, with numerous practical tables and curves indicate that this book is worth the reader's serious consideration. The text is so much worth while that the volume deserves a permanent place in the engineer's library. The thirty-sixth edition of this work contains 48 more pages than that of 1922 and new chapters have been added entitled "Mechanical Stokers," twenty pages, and "Pulverized Coal," four pages. Many new tables have been included, and some of the chapters increased, as a result of later practice. Many chapters have been shortened. Everywhere is evidence that painstaking and thorough revision has been carried out, so that the new edition will represent not only up-to-date practice but, if anything, a more comprehensive volume for its size than the preceding. The subject of steam boilers and their application, particularly the water-tube type, is taken up in a number of interesting chapters. The early history and outline of developments as well as the requirements of steam boilers and the evolution of the present types are described in a lucid and well-illustrated manner. Heat, steam making, properties of boiler-feed water and methods of feeding are treated satisfactorily and briefly. Numerous tables, corresponding to those of everyday use, are included, together with formulas, methods of using testing apparatus, etc. Chapters on boiler efficiency, operation, brickwork, piping data, etc., are also included.

Der Schiffsmaschinenbau. Erster band. By G. Bauer. Published by R. Oldenbourg, München and Berlin. 1923. Printed in German, 156 pages. Paper 8x11 in. 793 illustrations and 70 tables. Price $8.14.

United States Government Leather Belting Specifications — Circular 148, Bureau of Mines, Washington, D. C. The Federal Specifications Board have extended this investigation and now issue Standard Specifications No. 37, on Leather Belting. The greater part of the requirements has been standard for a number of years, but the tests called for are quite comprehensive and embody a number of new features. The board calls for tensile-strength tests, cracking tests, piping tests, waterproofing tests and chemical tests, including those of stuffing content, mineral acid and glucose. While these specifications were drawn up with the view of obtaining uniformity in the several government departments, firms who purchase much belting will find the specifications of advantage.

Coming Conventions

American Association for the Advancement of Science. Burton E. Livingston, Smithsonion Institution Bldg., Washington, D. C. Seventy-fifth anniversary meeting at University of Cincinnati, Cincinnati, Ohio, Dec. 27-Jan. 2.

American Engineering Council of the F.A.E.S. L. W. Wallace, 24 Jackson Place, Washington, D. C. Meeting at Washington, D C., Jan. 10-11.

American Institute of Chemical Engineers. J. C. Olsen, Polytechnic Institute, Brooklyn, N. Y. Winter meeting at the New Willard Hotel, Washington, D. C., Dec. 5-8.

American Institute of Electrical Engineers. F. L. Hutchinson, 29 West 39th St., New York City. Midwinter convention at Philadelphia, Feb. 4-8.

American Society of Heating & Ventilating Engineers. C. W. Obert, 25 West 39th St., New York City. Annual meeting at New York City, Jan. 22-25.

American Society of Mechanical Engineers, Calvin W. Rice, 29 West 39th St., New York City. Annual meeting at New York City, Dec. 3-6.

American Society of Refrigerating Engineers, William H. Ross, 35 Warren St., New York City. Nineteenth Annual Convention at Hotel Astor, New York City, Dec. 3-5.

American Society of Safety Engineers. Genevieve S. Wood, 29 West 39th St., New York City. Annual meeting at New York City, Jan. 18.

Electric Power Club, S. N. Clarkson, 900 B. F. Keith Bldg., Cleveland. Fall meeting at French Lick Springs Hotel, French Lick, Ind. Nov. 19-22.

Engineering Institute of Canada. Fraser S. Keith, 176 Mansfield St., Montreal, Canada. Annual General Meeting at Montreal, Quebec, Jan. 22—Ottawa, Ontario, Jan. 23-24.

Kansas Engineering Society. C. V. Waddington, Kansas Gas & Electric Co., Wichita, Kans. Convention at Wichita, Kans., Dec. 11-13.

National Association of Leather Belting Manufacturers. George H. Blake, P. O. Box 859, City Hall Station, New York City. Annual meeting at Hotel Pennsylvania, New York City, Nov. 21.

National Association of Practical Refrigerating Engineers; Ed. H. Fox, 5707 W. Lake St., Chicago. Fourteenth Annual Convention at Memphis, Dec. 12-16.

National Electric Light Association —Southeastern Division. Charles A. Collier, Georgia Railway & Power Co., Atlanta, Ga. Meeting at Hillsboro Hotel, Tampa, Fla., Nov. 19-22.

National Exposition of Power and Mechanical Engineering; Charles F. Roth, Room 1102, Grand Central Palace, New York City. Power Show, Dec. 3-8.

National Marine Engineers' Beneficial Association. Geo. A. Grubb, 313 Machinists' Bldg., Washington, D. C. Annual Convention at Franklin Square Hotel, Washington, D. C. Jan. 21-26.

Society of Automotive Engineers, Coker F. Clarkson, 29 West 39th St., New York City. Annual meeting at General Motors Bldg., Detroit, Mich., Jan. 22-25.

Society Affairs

The Cincinnati-Liberty Section No. 15, N.A.S.E., Cincinnati, Ohio, will listen to a paper by H. F. Mitch on "Recording Instruments," on Nov. 24.

The Salt Lake City, Utah Section of the A.S.M.E. will hear Dr. Bonner of the University of Utah speak on "Coal Storage," at the Chamber of Commerce on Nov. 21.

The Denver Section of the A.S.M.E. will hear H. B. Carpenter of the Colorado Fuel & Iron Co. on "By-Product Coke Oven Operations," at the Hotel Metropole on the evening of Nov. 23.

The Boston Section of the A.I.E.E. has as the subject for its Dec. 11 meeting, "Common Sense and Mathematics in Engineering," by R. E. Doherty, of the General Electric Co., Schenectady, N. Y.

The Seattle Engineers' Club will have a talk on "Mechanical Equipment of the Steamship, H. F. Alexander," by J. C. Gilmour, Port Engineers of the Pacific Steamship Co., and "Power Plants on United States Scout Cruisers (Omaha Class)."

The Metropolitan Section of the A.S.M.E. will hear two papers on "Pulverized Coal," at its meeting on Nov. 20. H. D. Savage, of the Combustion Engineering Co., and F. A. Scheffler, of the Fuller Engineering Co., will be the speakers.

The A.S.M.E. Spring Meeting will be held in Cleveland, Ohio, May 19-22, 1924. The papers to be presented at this meeting must be in the hands of the committee on meetings and program by March 1, 1924, and authors must be selected by Jan. 15, 1924.

Personal Mention

General Guy E. Tripp, Chairman of the board of the Westinghouse Electric and Manufacturing Co. has had conferred upon him by Japan the second degree order of the Sacred Treasurer in recognition of his activities in assisting Japanese officials in rebuilding the area devastated during the recent earthquake.

Emile Prat, of 64 rue de Miromesnil, Paris, France, who has been visiting in New York City, has returned home very well pleased with his sojourn in America. He wishes to extend his thanks to those who were interested in his subject of the preheating of air and water for boilers by the use of flue gases, and to those who were interested in forming a Société Franco-Americaine for engineers.

Business Notes

Rowland D. Burns, 39 Cortlandt St. now represents the Locke Regulator Co., of Salem, Mass., in New York City.

The Lombard Governor Co., Ashland, Mass., manufacturer of oil engines, has established a New York office at 30 Church St. with W. Merton Rice, formerly connected with Henry J. Gielow, Inc., and Cox & Stevens, in charge of it.

Dean Brothers Co., manufacturer of pumping machinery, 323 West 10th St., Indianapolis, announces that R. T. Gordon has been appointed district manager of the New York City office, which is at 141 Broadway.

The Galion Iron Works & Manufacturing Co. has recently arranged an industrial division of its company, which will be entirely devoted to coal handling equipment and will be in charge of Eugene Smith, with Edwin Fitts, consulting engineer, associated with him.

Trade Catalogs

Pipe, Wrought—National Tube Co. Booklet, "Seven Wonders of Wrought Pipe," describes the qualities of this piping.

Coupling, Flexible—De Laval Steam Turbine Co., Trenton, N. J. Catalog showing the construction and dimensions of this flexible coupling.

Grates—McClave-Brooks Co., Scranton, Pa. Catalog showing shaking and cut-off grates, type 2-A, for all grades of bituminous, semi-bituminous, lignite and bagasse.

Motors, Electric—Reliance Electric & Engineering Co., Cleveland, Ohio. Booklet, "Electric Motors, How to Choose and Use Them," gives valuable information in concise form.

Stones, Commutator — Martindale Electric Co., Box 35, Lakewood Branch, Cleveland, Ohio. Leaflet "Imperial Commutator Stones," describes these stones and the grinding tool for them. Prices are listed.

Gas Analyzers—Jos. W. Hays Corp. This catalog shows, by means of descriptions and numerous illustrations, the working of these analyzers and contains a table showing possible savings of CO_2 and fuel losses.

Stoker—McClave-Brooks Co., Scranton, Pa. Catalog describing the anthracite type M-A stoker for all grades of coke breeze and mixtures, is well illustrated and forms a part of the complete loose leaf catalog gotten out by this company.

Instruments, Steam Plants — C. J. Tagliabue Manufacturing Co., 88 Thirty-third St., Brooklyn, N. Y. Catalog 64-A, "Tag Steam Plant," contains information about temperature and pressure controllers, thermometers for all uses, gages and recording instruments. It is amply illustrated and filled with convenient data.

Fuel Prices

BITUMINOUS COAL

The following table shows the trend of the spot steam market in various coals (mine run bases, f.o.b. mines):

Coal	Market Quoting	Nov. 5, 1923	Nov. 12, 1923
Pool 1,	New York	$2.75@3.25	2.75@3.25
Smokeless,	Columbus	2.00@2.65	2.10@2.65
Clearfield,	Boston	1.70@2.40	1.70@2.60
Somerset,	Boston	2.00@3.00	4.25@4.35
Kanawha,	Columbus	1.75@2.00	1.75@2.00
Hocking,	Columbus	1.75@2.00	1.75@2.00
Pittsburgh No. 8	Cleveland	1.80@1.90	1.85@1.95
Franklin, Ill.	Chicago	2.25@3.00	2.25@3.00
Central, Ill.	Chicago	2.00@2.25	2.00@3.25
Ind. 4th Vein,	Chicago	2.50@2.75	2.50@2.75
West Ky.,	Louisville	1.50@1.85	1.50@1.85
S. E. Ky.,	Louisville	1.50@2.25	1.50@2.25
Big Seam,	Birmingham	1.75@2.15	1.75@2.15

New York—Nov. 15, light oil, tank car lots, 28@34 deg. Baumé, 3⅝c. per gal., 36@40 deg. 3¾c. per gal., f.o.b. Bayonne, N. J.

St. Louis—Nov. 6, tank-car lots, f.o.b. St. Louis; 24@26 deg., $1.65 per bbl.; 26@28 deg., $1.75; 28@30 deg., $1.80; 32@36 deg., gas oil, $1.95@2.00; 36@40 deg., distillate, $2.20@2.25 per bbl.

Pittsburgh—Oct. 30, f.o.b. local refinery, 30@34 deg., fuel oil, 5¼c. per gal.; 36@40 deg., fuel oil, 5⅝c.; 34 deg., neutral 8c. per gal.

Dallas—Nov. 9, f.o.b. local refinery, 26@30 deg., $1.35 per bbl.

Cincinnati—Nov. 13, tank-car lots, f.o.b. local refinery, 26@30 deg. Baumé, 5c. per gal.; 30@32 deg., 5¼c.; 38@42 deg., distillate, 6¼c. per gal.

Philadelphia—Nov. 9, 26@28 deg., $1.68@1.722 per bbl.; 16@20 deg., $1.64@1.68; 14@16 deg., $1.45@1.515.

Chicago—Nov. 9, 24@26 deg., $1.72 per bbl.; gas oil, 32@36 deg., $1.92 per bbl.

Boston—Nov. 12, tank-car lots, f.o.b. heavy oil, 12@14 deg. Baumé, 4¼c. per gal.; light oil, 28@32 deg. Baumé, 6¼c. per gal.

New Plant Construction

PROPOSED WORK

Ark., Gillett—The Youngblood Ice Co., A. L. Youngblood, Pres., is in the market for a 25 hp. oil engine to furnish light and power for ice plant.

Calif., Livermore — The United States Veterans' Bureau, Arlington Bldg., Wash., D. C., will receive bids until Dec. 22 for the construction of infirmary; administration, mess, kitchen and recreation buildings; nurses' and attendants' quarters; garage; storehouse; boiler house and laundry; occupational-therapy building, etc., including mechanical equipment. Separate bids will be received for building construction; central heating, including boiler plant and heating equipment for all buildings; water supply, sewers, etc.; electrical equipment; elevators; refrigerating plant; stack. Noted Sept. 25.

Calif., Los Angeles—The Southwest Trust & Savings Bank, 565 South Spring St., is having preliminary plans prepared for the construction of a 12 story bank and office building on Main and Commercial Sts. Dodd & Richards, Brack Shops Bldg., Los Angeles, Archt.

Calif., Stockton—Davis-Hellar-Pearce Co., Archts., Weber and California Sts., are receiving bids for the construction of college buildings, including power house, for the College of the Pacific. Estimated cost $600,000. Equipment detail not reported. Noted Aug. 21.

Calif., Turlock—The Turlock Irrigation Dist. received the lowest bids for the construction of high line main canal and cross ditch enlargement, also Ceres main canal enlargement from C. F. Tullock, Merced, $7,375 and $6,840 respectively; twenty four drainage pumping plants from the Byron Jackson Pump Mfg. Co., 55 New Montgomery St., San Francisco, $15,488. No bids were received for concrete lining and transmission lines and this work will be done by day labor, estimated cost $105,252 and $350,000 respectively. Noted Oct. 23.

Conn., Berlin—The Donnelly Brick Co. will soon award the contract for the construction of a 1 story, 60 x 160 ft. power house. Estimated cost $40,000. M. J. Unkelbach, Main St., New Britain, Archt. Noted Nov. 6.

Conn., Darien—The Cherry Lawn School is having plans prepared for the construction of a sewage disposal plant and a central heating plant, estimated cost $200,000; also for a swimming pool, including pumps and filters, cost will exceed $25,000. W. R. Steinmetz, Greely Bldg., Stamford, Archt.

Fla., Cocoanut Grove—The Seaview Hotel, Inc., Miami, C. D. Briggs, Pres., is having plans prepared for the construction of an 8 story, 231 x 322 ft. hotel on Bayshore Rd., here. Estimated cost $2,000,000. J. C. Gault, Miami, Archt. Equipment detail not reported.

Ind., Indianapolis — The Bd. of County Comrs., Court House, will receive bids until Dec. 10 for the construction of a 1 and 2 story, 100 x 120 ft. administration building, 1 story, 30 x 50 ft. power plant, etc., including heating plant, for the Colored Orphans' Home on 25th and Keystone Sts. Estimated cost $140,000, heating plant $17,000. D. Graham, Hume-Mansur Bldg., Indianapolis, Archt. Bids will also be received until Dec. 11 for refrigerating plant.

Ia., Manson—The Manson Light, Heat & Power Co. plans to install new equipment and to change its system from direct to alternating current within a year, under the terms of a new 25 year franchise.

Ia., Wyoming—The city voted $12,500 bonds for waterworks extension and $35,000 for an electric light plant.

Kan., Olathe—The state, A. Doerr, Business Mgr., Topeka, is in the market for two 150 hp. steam boilers for heating plant of State School for Deaf, here.

Kan., Pittsburgh — The Hull & Dillon Packing Co., A. Dillon, Purch. Agt., plans a raw water ice plant in connection with present meat packing plant.

Ky., Louisville—The Louisville Ry. Co., 314 West Jefferson St., will receive bids until Dec. 3 for the construction of a 4 story, 76 x 490 ft. office building and interurban station on 2nd and Liberty Sts. Estimated cost $1,250,000. Joseph & Joseph, Francis Bldg., Archts. Equipment detail not reported. Noted Sept. 25.

Ky., Paintsville—The City Council plans extension to waterworks and substitution of electrically operated pumping machinery for present steam plant. Estimated cost $50,000.

La., Erath—The Mayor and Bd. of Aldermen, City Hall, will receive bids until Nov. 27 for the construction of an electric light and power pole line, consisting of furnishing and installing one 2,300 volt, 3 phase, 60 cycle a.c. pole line from Abbeville, electric lighting and power distributing system, transformers, etc., here.

Miss., Jackson—The Jackson Infirmary, c/o N. C. Womack, will receive bids until Nov. 22 for an oil burning boiler, 48 in., 8 sections, for about 2,355 ft. of radiation.

Mo., Kansas City—The Marshall Mfg. Co., 912 Grand Ave., is in the market for a 25 hp. motor.

Mo., Kirksville—The city is having preliminary plans prepared for the construction of an electric light plant and distribution system. Cost between $27,000 and $35,000. Black & Veatch, Mutual Bldg., Kansas City, Engrs.

Mo., Potosi—The city is having preliminary plans prepared for the construction of a waterworks pumping plant, stand pipe, distribution mains, etc. Estimated cost $30,000. F. W. Graham, 5244 Cote Brilliante, St. Louis, Engrs.

Mo., St. Joseph—The St. Joseph Light & Power Co. plans the construction of a power plant and the installation of equipment. Estimated cost $2,000,000. Engineer or architect not selected.

Mo., St. Louis—A local syndicate, c/o R. P. Roby, 300 Odd Fellows Bldg., is having preliminary plans prepared for the construction of a 12 story, 100 x 120 ft. office building on Delmar St. Estimated cost $1,500,000. Mauran, Russell & Crowell, Chemical Bldg., St. Louis, Archts.

N. J., Riverton—The borough is having plans prepared for the construction of pipe line, disposal plant, Imhoff tanks, sprinkling filters, filter beds, pumping station and equipment, including centrifugal pumps, electrically operated. Estimated cost $85,000. Sherman & Sleeper, Camden, Engrs.

N. Y., Brooklyn—The Superintendent of the Bd. Educ., 500 Park Ave., New York, will receive bids until Nov. 26 for the construction of P. S. 202 on Hegeman and Atkins Ave., here. W. H. Gompert, Flatbush Ave. Extension and Concord St., Brooklyn, Archt. Equipment detail not reported.

N. Y., East Greenwich—The Salem Power & Light Co., Salem, J. P. O'Brien, Pres., is in the market for generator, balancer set, governor, etc., for hydro-electric plant, here. Estimated cost $10,000.

N. Y., Holley—R. Sarmsworth is in the market for refrigeration equipment for cold storage plant, to replace that which was destroyed by fire.

N. Y., New York—N. C. Partos, c/o L. A. Abramson, Engr. and Archt., 46 West 46th St., is having plans prepared for the construction of a 12 story apartment house on East 97th St. Equipment detail not reported.

N. Y., New York—The Supt. of Bd. Educ., 500 Park Ave., will receive bids until Nov. 26 for the construction of P. S. 99 on Exterior St. Estimated cost $1,500,000. W. H. Gompert, Flatbush Ave., Extension and Concord St., Brooklyn, Engr. and Archt. Equipment detail not reported. Noted Nov. 13.

N. Y., Ogdensburg—The Hull Steel Fdrys. Co., Hull, Que., manufacturer of railroad equipment, A. H. Coplan, Pres., is having plans prepared for the construction of a steel mill, including a 100 x 600 ft. main building, pattern department, machine shop, power house, hospital, storage and office buildings, etc., here. Estimated cost $1,000,000. Private plans.

N. Y., Palmyra — The State Comr. of Canals, Albany, received low bid for completing the construction of power station at Lock 29, here, Cont. 193, from T. H. Ryan, Inc., Buffalo, $59,529. Noted Oct. 16.

N. Y., Vischer Ferry—The State Comr. of Canals, Albany, received low bid for the construction of superstructure of power house at the dam on the canalized Mohawk River, here, Cont. B-4, from the McCann Bldg. Co., Sheridan Ave. and Dove St., Albany, $143,567. Noted Oct. 23.

N. C., Greensboro—The North Carolina College for Women, J. I. Forrest, Pres., will receive bids until Dec. 4 for the construction of three dormitories, physical education building, new wing, pavillion and dining hall, including heating and electrical equipment, etc. Cost will exceed $600,000. H. Barton, Greensboro, Archt.

N. C., Morganton—The Bldg. Com. of the State Hospital, Charlotte, will receive bids until Nov. 24 for the construction of (1) power house chimney, machinery foundations, concrete pipe tunnels, etc.; (2) radial brick stack; (3) hand stokers for four 300 hp. boilers, here. Wiley & Wilson, Lynchburg, Va., Consult. Engrs.

Ohio, Mount Sterling—The city plans the construction of a power and water plant, to replace the one which was recently destroyed by fire. Estimated cost $50,000. Engineer or architect not selected.

Okla., Purcell—The Atchison, Topeka & Santa Fe Ry., 179 West Jackson St., Chicago, is having plans prepared for the construction of an addition to its power plant, here. Estimated cost $30,000. Reinhart & Donovan, 803 Traders Natl. Bldg., Oklahoma City, Engrs.

Ore., La Grande—The city is having plans prepared for the construction of a 15 mi. pipe line, estimated cost $270,000; twin reservoirs, $35,000; pumping plant, $25,000. F. B. Hayes, Engr., is interested in prices of pumps, boosters and engines, capacity 2,000,000 to 3,000,000 g.p.d. against a 150 ft. head, electric drive.

Pa., Dawson—The Corrado Coal Co., Fort Hill, is in the market for coal handling equipment, tipple machinery and boiler house equipment.

Tenn., Memphis — W. W. Ahlschlager, Archt., 65 East Huron St., Chicago, is receiving bids for the construction of a 12 story, 170 x 324 ft. hotel including steam heating system, on Union, 2nd and 3rd Sts., here. Estimated cost $3,000,000. Owner's name withheld.

Tex., Beaumont — The Eastern Texas Electric Co. is having plans prepared for the construction of a 2 story, 80 x 100 ft. power plant and distribution station on Travis St. Cost between $40,000 and $50,000. Private plans.

Tex., Houston—The City Comn. is receiving bids for the construction of a sewer line and sewerage pumping plant in the Eastwood section. Estimated cost $33,500. J. C. McVea, Engr.

Tex., San Antonio—S. L. Jeffers, 1 Bushman Pl., is having preliminary plans prepared for the construction of a 100 x 140 ft. hotel on Ave. E and 3rd St. Estimated cost $400,000. H. T. Phelps, 519 Hicks Bldg., Archt. Equipment detail not reported.

Tex., San Benito—The Valley Electric & Ice Co., is having surveys made for the construction of a central power plant and transmission lines to principal points in the Rio Grande Valley. Private plans.

Wash., Centralia—The City Council plans to build a pumping and filtration plant, capacity 1,500,000 g.p.d. on the Skookumchuck River. Other units will be constructed later to increase capacity to 6,000,000 gal. R. A. Welsh, Engr.

Wash., Tacoma—The Citizens Hotel Corp., Inc., 610 Tacoma Bldg., has had plans prepared and will receive bids Dec. 7 for the construction of a 12 story hotel. Estimated cost $1,500,000, including site. W. L. Stoddart, 50 East 41st St., New York, N. Y., Archt. Equipment detail not reported. Noted June 20, 1922.

W. Va., Huntington—The Fesenmeier Packing Co., West 14th St., is having plans prepared for the construction of an addition to its ice and cold storage plant. Estimated cost $40,000. E. F. Glaser, 2841 Madison Rd., Cincinnati, Ohio, Archt.

Wis., Fond du Lac—The H. & H. Cafeteria Co., 105 South Main St., E. C. Halverson, Mgr., is in the market for an ice machine and refrigeration machinery, brine type.

Ont., Oshawa—The town is in the market for a 1,000 g.p.m., 3 stage centrifugal pump to work against a head of 340 ft., direct connected to a 60 cycle, 550 volt squirrel cage induction motor. W. C. Smith, Town Hall, Engr.

Ont., Windsor—The Essex Border Utilities Comn. is receiving bids for the construction of filtration beds and reservoir. Bids will be received about Jan. 1 for the construction of pump house, pipe lines, pumping and filtration equipment. Estimated cost $150,000. J. C. Keith, c/o owner. Engr. Noted Sept. 4.

Que., Chicoutimi—Price Bros. & Co., Ltd., 56 St. Pierre St., Quebec, are in the market for equipment for hydro-electric power house on the Chicoutimi River, here.

Que., Lachine Rapids — The Montreal Light, Heat & Power Co., 83 Craig St., W., Montreal, plans to build a power house to develop 100,000 hp., here. Estimated cost $2,000,000. R. M. Wilson, Ch. Engr. The owner is in the market for equipment.

CONTRACTS AWARDED

Ia., Albia—The city awarded the contract for the construction of an impounding reservoir, filtration plant and pumping station to A. Phelps & Son, Knoxville. Estimated cost $50,000. Noted Oct. 2.

Ia., Cedar Rapids—The Iowa Ry. & Light Co., Bowes Bldg., awarded the contract for the construction of a 1 story, 45 x 100 ft. addition to power plant to T. Stark & Co., 118 South 11th St., Cedar Rapids. Estimated cost $365,000.

Ky., Powderly—The Illinois Central R.R., 135 East 11th Pl., Chicago, awarded the contract for the construction of a water treating plant and pumping station, here, to J. E. Nelson & Son, 3240 South Michigan Ave., Chicago. Estimated cost $40,000.

Minn., Duluth—W. Schroeder & Sons, 86 Michigan St., Milwaukee, awarded the contract for the construction of foundation for a 16 story, 125 x 140 ft. hotel on Superior St. and 3rd Ave. E., here, to Jacobson Bros., 410 Columbia Bldg., Duluth. Bids for the general contract, including steam heating system, will be opened about Jan. 1, 1924. W. Tullgren & Sons Co., 425 East Water St., Milwaukee, Archts. Noted Sept. 18.

Minn., Minneapolis—The Northern States Power Co., 15 South 5th St., will build a 4 story, 56 x 57 x 130 ft. substation at 1057 1st Ave. N., by day labor. Estimated cost $90,000.

N. Y., Buffalo—The Iroquois Natural Gas Co., Iroquois Bldg., awarded the contract for the construction of an extension to boiler house to the Turner Constr. Co., 11 Goodell St., Buffalo.

N. Y., Linoleumville (Staten Island P. O.) —The Amer. Linoleum Mfg. Co. awarded the contract for the construction of a power plant to the W. G. Hudson Corp., 50 Church St., New York. Estimated cost $100,000.

N. Y., New York—The Bd. Educ., 500 Park Ave., awarded the contract for the construction of James Monroe High School, Boynton Ave., to the T. A. Clarke Co., 122 Livingston Ave., Brooklyn. $2,145,000 Equipment detail not reported. Noted Oct. 16.

N. Y., New York—The Commissioner of Water Supply, Gas & Electricity, Municipal Bldg., awarded the contract for furnishing and installing electric valve operating equipment to the Fox Reynolds Co., 81 East 125th St., $25,500. Noted Nov. 6.

Okla., Sulphur—The city, F. Gofford, Clk., awarded the contract for two motor driven centrifugal pumps, concrete reservoir, tower and pipe line to W. B. Scott, Sulphur. Estimated cost $45,000.

Okla., Tulsa—The City Water Comn., A. J. Rudd, Chn., awarded contracts for the construction of a portion of the Spavinaw water supply project as follows: (Contract 14) Mohawk pumping station to E. J. Merkle, 1733 Walnut St., Kansas City, Mo., $220,000; (15) concrete chimney to the J. V. Boland Constr. Co., Chemical Bldg., St. Louis, Mo., $4,984; (16) 30 in. force main, hammer weld steel, to the H. L. Cannady Constr. Co., Tulsa, $230,263.60. Noted Oct. 2.

Pa., Morrisville—D. C. Cook, 25 Fair St., Trenton, N. J., awarded the contract for the construction of a 50 x 75 ft. ice plant on Bridge St., here, to E. D. LaRue, 53 South Hermitage St., Trenton, N. J. Estimated cost $100,000. The owner will install ice making machinery, capacity 50 ton per day.

Vt., East Arlington—The Twin State Gas & Electric Co., Bennington, will build a 20 mi., 23,000 volt, 3 phase, 60 cycle transmission line from Bennington to Arlington, also a transformer station, etc., here. The distribution system will be about 12 mi. additional. The owner will purchase transformers, breakers, poles, wire, insulators, etc.

Wash., Wenatchee — The Puget Sound Power & Light Co., Stuart Bldg., Seattle, awarded the contract for the construction of a transforming station, here, to the Stone & Webster Eng. Corp., 7th and Olive Sts., Seattle. Estimated cost $120,000. This is one of the final links in a chain of electrical equipment which this company has built to bring 30,000 hp. from the White River plant into north central Washington.

Wis., Milwaukee—The city, Central Bd. of Purchases, awarded the contract for a feed water heater for Riverside pumping station to Cochrane Eng. Co., 53 West Jackson Blvd., Chicago, $11,183. Noted Oct. 30.

Wis., West Bend—The Atlantic & Pacific Products Co. awarded the contract for the construction of foundation of condensery, boiler house, etc., to the West Bend Constr. Co. Estimated cost $25,000.

Ont., Toronto — The Toronto Electric Comrs., Yonge and Shuler Sts., awarded the contract for the construction of a 1 story, 60 x 80 ft. electric substation on Danforth and Morton Sts., to the Jackson Lewis Co., Ltd., Ryrie Bldg., Toronto. Estimated cost $300,000. The owner will purchase transformers, lightening arresters, oil switches, wire, buses, switchboards, etc. Noted Oct. 16.